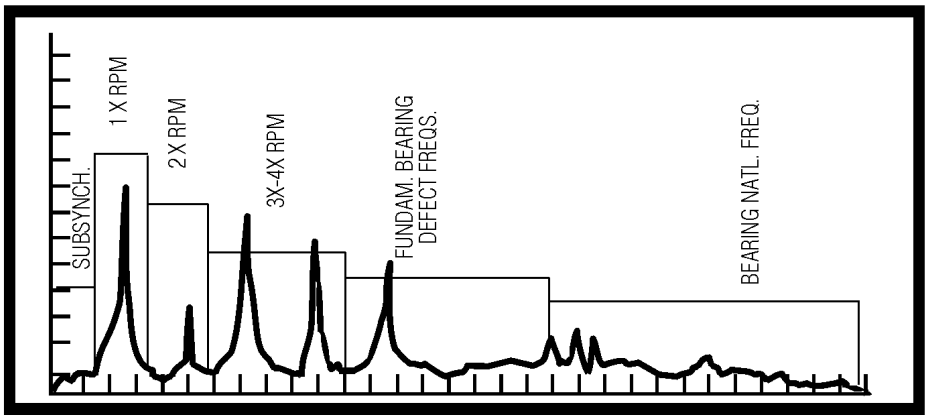
ANALYSIS II

CONCENTRATED VIBRATION SIGNATURE ANALYSIS AND RELATED CONDITION MONITORING TECHNIQUES

Metric Version



Authored By:
Mr. James E. Berry, P.E.
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SPECIALISTS IN PREDICTIVE MAINTENANCE, MACHINERY DIAGNOSTICS, AND VIBRATION REDUCTION

PREDICTIVE MAINTENANCE AND VIBRATION SIGNATURE ANALYSIS II

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SPECIALISTS IN PREDICTIVE MAINTENANCE, MACHINERY DIAGNOSTICS, AND VIBRATION REDUCTION

We thank you for the opportunity to assist you in development of your vibration analysis and condition monitoring skills. As you can see from the length and breadth of the information within this text, many variables go into application of the material and so much depends upon the knowledge and judgement of the analyst and/or technician. The author and his company and /or companies can assume no responsibility for results in spite of many years of successful and trouble-free personal application.

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ANALYSIS II

TABLE OF CONTENTS

Chapter **Topic** **Page**

• Recommended Periodicals for those Interested in Predictive Maintenance i

1. Seminar Overview 1-1

2. Brief Review of "ANALYSIS I" Seminar Topics

2.0 Introduction 2-1
2.1 What is Vibration and How Can it be Used to Evaluate Machinery Condition? 2-1
2.11 Introduction 2-1
2.12 What is Vibration Frequency and How Does it Relate to a Time Waveform? 2-2
2.13 What is Vibration Amplitude? 2-4
2.131 What is Vibration Displacement? 2-4
2.132 What is Vibration Velocity? 2-4
2.133 What is Vibration Acceleration? 2-5
2-14 What is Vibration Phase? 2-5
2.141 How to Read Phase on CRT or RTA Screens 2-6
2.142 Phase Relationship of Acceleration, Velocity & Displacement Time Waveforms 2-8
2.15 What is a Vibration Spectrum (Also Called an "FFT" or "Signature")? 2-8
2.16 Difference Between RMS, Peak and Peak-To-Peak Vibration Amplitude 2-10
2.17 When to Use Displacement, Velocity, or Acceleration 2-12
2.171 What is the Advantage of Using Velocity? 2-17
2.18 How Much is Too Much Vibration? 2-17
2.2 Overview of the Strengths and Weaknesses of Typical Vibration Instruments 2-20
2.21 Introduction 2-20
2.22 Instrument Comparisons 2-20
2.23 General Capabilities of Each Vibration Instrument Type 2-24
2.231 Overall Level Vibration Meters 2-24
2.2311 Drawbacks in Measuring only Total or Overall Vibration 2-24
2.232 Swept-Filter Analyzers 2-24
2.233 FFT Programmable Data Collectors 2-25
2.234 Real-Time Spectrum Analyzers 2-25
2.235 Instrument Quality Tape Recorders 2-26
2.3 Overview of Vibration Transducers and How to Properly Select Them 2-32
2.31 Introduction 2-32
2.32 Types of Vibration Transducers and Their Optimum Applications 2-33
2.321 Accelerometers 2-33
2.322 Velocity Pickups 2-38
2.323 Noncontact Eddy Current Displacement Probes 2-42
2.324 Shaft Contact Displacement Probes 2-46
2.3241 Shaft Sticks 2-46
2.3242 Shaft Riders 2-47

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2.33 Selection Criteria for Transducers . . . 2-48
2.34 Mounting of Transducers (Accelerometers) . . . 2-50
2.341 Transducer Mounting Applications . . . 2-50
2.4 Understanding Vibration Phase and Its Applications . . . 2-53
2.41 Introduction . . . 2-53
2.42 How to Make Phase Measurements . . . 2-53
2.43 Using Phase Analysis in Vibration Diagnostics . . . 2-55
2.431 Evaluating Axial Motion of a Bearing Housing to Reveal a Possible Cocked Bearing or a Bent Shaft . . . 2-55
2.432 Phase Behavior Due to Unbalance . . . 2-56
2.433 Phase Behavior Due to Looseness/Weakness . . . 2-56
2.434 Phase Behavior Due to Misalignment . . . 2-58
2.435 Using Phase Analysis to Find the Operating Deflection Shape of a Machine and Its Base . . . 2-59
Appendix - Specifications for Various Transducers From a Variety of Manufacturers . . . 2-63

3. Principles of Digital Data Acquisition and FFT Processing for Spectral Analysis

3.0 Introduction . . . 3-1
3.1 FFT Properties . . . 3-1
3.11 How Many Spectral Lines are There? . . . 3-3
3.12 What is the Spacing of the Lines? . . . 3-4
3.13 What is the Frequency Range of the FFT? . . . 3-4
3.2 Sampling and Digitizing . . . 3-4
3.3 Aliasing . . . 3-5
3.31 Aliasing in the Frequency Domain . . . 3-5
3.32 The Need For an Anti-Alias Filter . . . 3-5
3.33 The Need For More Than One Anti-Alias Filter . . . 3-5
3.34 Digital Filtering . . . 3-5
3.35 Formulas Used to Calculate t_{MAX} and F_{MAX} . . . 3-6
3.4 Window Selection . . . 3-7
3.41 The Need For Windowing . . . 3-7
3.42 What is Windowing? . . . 3-7
3.43 The Hanning Window . . . 3-10
3.44 The Uniform (Rectangular Window) . . . 3-12
3.45 The Flat Top Window . . . 3-12
3.5 Averaging . . . 3-13
3.51 RMS (Power) Averaging . . . 3-13
3.52 Linear Averaging and Synchronous Time Averaging . . . 3-13
3.6 Overlap Processing . . . 3-15
3.61 Example of Sampling Times With and Without Overlap Processing . . . 3-16
3.7 Understanding a Vibration Spectrum . . . 3-18
3.71 Effect of the Number of FFT Lines Used on Frequency Accuracy . . . 3-20
3.72 Effect of the Frequency Span Used on Frequency Accuracy . . . 3-23
3.73 Improving the Frequency Resolution with “Zoom”- Band Selectable Fourier Analysis . . . 3-25
3.74 Improving the Precision of the Spectrum by Frequency and Amplitude Interpolation . . . 3-29

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Chapter	Topic	Page
	3.75 Improving the Frequency Accuracy by Checking the Bandwidth	3-33
	3.76 Effect of Dynamic Range on Frequency and Amplitude Display	3-36
3.8	**What is Overall Vibration?**	3-42
	3.81 Digital (or Spectral) Overall Level	3-42
	3.82 Analog Overall Level	3-44

4. Introduction to Natural Frequency Testing and Instrumentation

4.0	**Introduction**	4-1
4.1	**Difference Between Natural Frequency, Resonance and Critical Speed**	4-1
4.2	**Change in Mode Shape with Higher Natural Frequencies**	4-5
4.3	**Impact/Impulse Natural Frequency Testing**	4-6
4.4	**Runup and Coastdown Natural Frequency Tests**	4-15
	4.41 Bode' Plots	4-15
	4.411 A Tracking Filter is Needed for Bode' Plots	4-15
	4.412 Explanation of a Bode' Plot	4-15
	4.413 Interpreting Unusual Bode' Plots	4-16
	4.42 Polar Plots	4-23
	4.421 Setting Up for Polar Plots	4-23
	4.422 Advantages of Polar Plots over Bode' Plots	4-23
	4.423 Comparison of Bode' and Polar Plots for Natural Freq. Testing	4-24
	4.424 Applying Polar Plots to Natural Frequency and Resonance Diagnostics	4-25
	4.425 Limitations of Polar Plots	4-25

5. Enhanced Vibration Diagnostics Using Cascade Diagrams

5.0	**Introduction**	5-1
5.1	**Diagnosis of Rotor Rub Problems**	5-1
5.2	**Diagnosis of Serious Oil Whirl and Oil Whip Problems**	5-5
5.3	**Diagnosis of Resonant Frequencies**	5-6

6. Use of Vibration Signature Analysis to Diagnose Machine Problems

6.0	**Use of Vibration Signature Analysis**	6-1
	TABLE 6.0 Illustrated Vibration Diagnostic Chart (Showing Typical Spectra & How Phase Reacts)	6-4
	6.01 Mass Unbalance	6-12
	6.011 Force Unbalance	6-15
	6.012 Couple Unbalance	6-15
	6.013 Dynamic Unbalance	6-16
	6.014 Overhung Rotor Unbalance	6-17
	1. Balancing Overhung Rotors by Classic Single-Plane Static-Couple Method	6-18
	2. Balancing Overhung Rotors by Classic Two-Plane Static-Couple Method	6-20
	6.015 Allowable Residual Unbalance & ISO Balance Quality Grade	6-21

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Chapter	Topic	Page
6.02	**Eccentric Rotors**	6-27
6.03	**Bent Shaft**	6-30
6.04	**Misalignment**	6-32
6.041	Angular Misalignment	6-35
6.042	Parallel Misalignment	6-36
6.043	Misaligned Bearing Cocked on the Shaft	6-37
6.044	Coupling Problems	6-37
6.05	**Machinery Failures Due to Resonant Vibration**	6-39
6.051	Identifying Characteristics of Natural Frequencies That Help "Give Them Away"	6-45
6.052	How Natural Frequencies Can Be Approximated For Overhung Rotors and Machines with Loads Supported Between Bearings	6-48
6.06	**Mechanical Looseness**	6-51
6.061	Type A - Structural Frame/Base Looseness (1X RPM)	6-51
6.062	Type B - Looseness Due to Rocking Motion or Cracked Structure/Bearing Pedestal (2X RPM)	6-55
6.063	Type C - Loose Bearing in Housing or Improper Fit Between Component Parts (Multiple Harmonics)	6-55
6.07	**Rotor Rub**	6-61
6.071	Partial Rub	6-63
6.072	Full Annular Rub	6-64
6.08	**Journal Bearing Problems**	6-68
6.081	Journal Bearing Wear and Clearance Problems	6-71
6.082	Oil Whirl Instability	6-73
6.083	Oil Whip Instability	6-75
6.084	Dry Whip	6-75
6.09	**"Tracking of Rolling Element Bearing Failure Stages Using Vibration Signature Analysis"**	6-76
6.091	Optimum Vibration Parameter For Bearing Problem Spectra (Acceleration, Velocity & Displacement)	6-79
6.092	Types of Vibration Spectra Caused By Defective Rolling Element Bearings	6-81
6.093	Typical Spectra For Tracking Failure Stages Through Which Rolling Element Bearings Pass	6-101
SCENARIO A.	4 Primary Failure Stages Through Which Most Rolling Element Bearings Pass	6-102
SCENARIO B.	Continued Deterioration of one Pronounced Fault on a Raceway	6-119
SCENARIO C.	Continual Wear Throughout the Periphery of one Raceway	6-119
SCENARIO D.	Development of a Serious Fault Frequency Acting as a Sideband Rather Than a Fundamental	6-122
SCENARIO E.	Condition Deterioration Ending Either with Severe Mechanical Looseness or the Bearing Turning on the Shaft	6-122
SCENARIO F.	Development of Excessive 1X RPM Modulation About Race Frequencies Ending Up with Multi-Harmonics	6-122

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Chapter	Topic	Page
6.094	Word of Warning Concerning Instruments and Transducer Mountings	6-126
	a. How 8-Bit Data Collectors Can Miss Potentially Serious Bearing Problems	6-126
	b. Impact of Transducer Mounting on Detecting Rolling Element Bearing Problems	6-130
6.095	Recommendations on When Rolling Element Bearings Should Be Replaced	6-134
TABLE 6.09B	When To Replace Rolling Element Bearings on Noncritical General Machinery Versus on Critical, Expensive Machinery	6-135
6.10	**Flow-Induced Vibration**	6-136
6.101	Hydraulic and Aerodynamic Forces	6-136
6.102	Cavitation and Starvation	6-139
6.103	Recirculation	6-139
6.104	Flow Turbulence	6-140
6.105	Surge	6-143
6.106	Choking	6-143
6.11	**Gear Problems**	6-144
6.111	Gear Tooth Wear	6-146
6.112	Significant Load Imposed on Gear Teeth	6-147
6.113	Gear Eccentricity and/or Backlash	6-147
6.114	Gear Misalignment	6-148
6.115	Cracked, Chipped or Broken Gear Teeth	6-148
6.116	Hunting Tooth Problem	6-149
6.12	**Electrical Problems**	6-151
6.121	Stator Problems	6-156
6.122	Eccentric Rotor (Variable Air Gap)	6-158
6.123	Rotor Problems	6-160
6.124	Thermal Bow Induced by Uneven Localized Heating of a Rotor	6-166
6.125	Electrical Phasing Problems (Loose Connectors)	6-166
6.126	Synchronous Motors (Loose Stator Coils)	6-169
6.127	DC Motor Problems	6-171
6.128	Torque Pulse Problems	6-173
6.13	**Belt Drive Problems**	6-175
6.131	Worn, Loose or Mismatched Belts	6-176
6.132	Belt/Sheave Misalignment	6-178
6.133	Eccentric Sheaves	6-178
6.134	Belt Resonance	6-179
6.135	Excesive Motor Vibration At Fan Speed Due to Motor Frame/Foundation Resonance	6-179
6.136	Loose Pulley or Fan Hub	6-179
6.14	**Beat Vibration Problems**	6-180

7. Proven Method for Specifying Both 6 Spectral Alarm Bands as well as Narrowband Alarm Envelopes using Today's Predictive Maintenance Software Systems

7.0	**Abstract**	7-1
7.1	**Introduction to Specifying Spectral Alarm Bands & Frequency Ranges**	7.2
7.11	Two Types of Spectral Alarm Bands	7-3

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Chapter	Topic	Page
7.12	Which Vibration Parameter to Use in Spectral Alarm Bands -Displacement, Velocity or Acceleration?	7-4
7.13	Review of Problems Detectable by Vibration Analysis	7-5
7.14	Specification of Overall Vibration Alarm Levels and Explanation of The Origin of Table II "Overall Condition Rating" Chart	7-13
7.15	Specification of Spectral Alarm Levels and Frequency Bands UsingTable III	7-14
7.151	Examples	7-22
7.16	Periodic Reevaluation of Spectral Alarm Band Setups on Each Family of Machines	7-29
7.161	Procedure for Evaluating the Effectiveness of Specified Overall Alarm Levels and Spectral Bands	7-31
7.162	EXAMPLE - "Statistical Analysis of Overall Vibration Velocity in 4 Client Power Plants Using the Procedure Recommended Above"	7-32
7.17	Conclusions	7-34
7.2	**How to Specify Narrowband Spectrum Alarms Using Statistical Alarm and Percent Offset Methods**	7-36
7.21	Introduction	7-36
7.22	What Narrowband Spectrum Alarms Are	7-37
7.23	Specifying the Narrowband Spectrum Alarm Limits	7-38
7.231	General Discussion	7-38
7.232	Generating Alarms When Setting Up a New Database	7-39
7.2321	Example - Setting Narrowband Spectrum Alarms for a Number of Belt-Driven Fans	7-40
7.233	Now for the Statistics	7-41
7.234	What About Unique Machines that Cannot be Comfortably Grouped Together?	7-45
7.24	Generating Alarm Values for a Pre-Existing Database	7-46
7.241	Specification of Narrowband Spectrum Alarms for Variable-Speed Machinery	7-47
7.25	Summary	7-48

8. Introduction to Lissajous Orbit Acquisition and Interpretation

8.0	**Introduction**	8-1
8.01	What is a Lissajous Orbit?	8-1
8.02	A Typical Setup for Generating Lissajous Orbits	8-2
8.03	Setting Up the Noncontact Pickups for Lissajous Orbits	8-2
8.04	Providing a Once-Per-Revolution Reference Pulse	8-2
8.05	The Oscilloscope - The Conventional Choice	8-3
8.06	Consideration of Pickup Location, Direction of Shaft Motion, and the Polarity of the Power Supply	8-3
8.07	Setting Up the Oscilloscope for Lissajous Orbits	8-3
8.08	Interpreting Lissajous Orbits With or Without "Blank" Spots	8-4
8.1	**Typical Lissajous Plots for Common Problems**	8-6
8.11	Unbalance	8-6
8.12	Misalignment	8-7
8.13	How Can It Be Determined Whether the Lissajous Orbit is Caused By Unbalance, Misalignment or Resonance?	8-9

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Chapter	Topic	Page
	8.14 Rotor Rubs	8-9
	8.141 Very Mild Rotor Rubs	8-9
	8.142 Heavy or Full Rubs	8-11
	8.143 "Hit and Bounce" Rubs	8-12
	8.144 Conclusions From Lissajous Orbits in Rotor Rub Diagnostics	8-12
	8.15 Oil Whirl	8-13
	8.16 Mechanical Looseness	8-13
	8.17 How Can it Be Determined Whether the Lissajous Orbit is From Mechanical Looseness, Rotor Rub or Oil Whirl?	8-14
	8.2 Applications of Lissajous Orbits Not Covered	8-14
9. Role of Spike Energy, HFD and Shock Pulse (SPM) & Specification of Their Alarm Levels at Various Speeds		
	9.1 Spike Energy and Shock Pulse	9-1
	9.2 High-Frequency Acceleration (HFD)	9-7
	9.3 Spike Energy Measurements	9-8
	9.4 High Frequency Enveloping and Demodulation Techniques	9-11
	9.41 IRD FAST TRACK gSE Spectrum	9-12
	9.5 Case Studies	9-18
10. Introduction to Vibration Isolation Mechanisms		
	Definition of Vibration Isolation	10-1
	Why are Isolators Needed?	10-1
	How Does Vibration Isolation Work?	10-4
	What is a Good Rule of Thumb for Specifying Proper Vibration Isolators?	10-4
	How Does the Amount of Isolator Damping Affect Isolator Performance?	10-5
	What are Some Typical Types of Isolators and How Does Their Performance Compare?	10-5
	Real-World Case History - "Provision of an Effective Isolation System to Prevent Transmission of Vibration into an Electron Microscope from a 2-Stage Reciprocating Air Compressor to be Installed on the First Floor Directly Beneath the Microscope Lab"	10-10
11. Introduction to Damping Treatments		
	Definition of Vibration Damping	11-1
	Types of Damping Treatments (Free Layer and Constrained Layer Damping)	11-8
12. Glossary		
13.* Real-World Case Histories		13-1

(Series of Case Histories based on actual experience will be included illustrating detection and correction of problems including unbalance, misalignment, looseness, rotor rub, sleeve bearing problems, rolling element bearing problems, gear problems, electrical problems, cavitation, belt-drive problems, beat vibration, soft foot, etc.).

*** NOTE:** A Table of Contents for "Real-World Case Histories" is found at the beginning of Section 13.

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SAMPLE

TECHNICAL ASSOCIATES PRE-SEMINAR QUESTIONNAIRE

SAMPLE

YOUR NAME: JOHN W. DOE
TITLE: Technician
COMPANY: XYZ Chemicals Inc.
LOCATION: Americana, TX
PHONE: (512)222-1233 EXT: 765
EMAIL: _____ FAX: _____

SEMINAR DATES: April 8-11, 1997
SEMINAR LOCATION: Myrtle Beach, SC
SEMINAR LEVEL (CIRCLE) ENTRY (I) II III IVD ADV
VIBRATION EXPERIENCE: YRS. ____ MOS. 6
SPECTRAL ANAL. EXPER: YRS. ____ MOS. 0

PLEASE COMPLETE THE FOLLOWING QUESTIONNAIRE, THIS INFORMATION IS FOR USE IN THIS CLASS AND TO HELP US PLAN FUTURE SEMINARS: ***(SEE SAMPLE PRE-SEMINAR SURVEY ENCLOSED)***

A. TYPE OF INDUSTRY AND PRODUCT MANUFACTURED/SERVICES RENDERED BY YOUR COMPANY:

AUTOMOTIVE		CHEMICAL	✓	CONSUMER PROD.	
CONSULTANT		FOOD/BEVERAGES		FABRICA. METAL	
FIBERGLASS		HVAC		INSTRUM./S'WARE	
MINING		PETROLEUM		PHARMACEUTICAL	
PULP & PAPER		REPAIR		SYNTHETIC FIBER	
TEXTILES		TOBACCO		UTILITIES	
VIB. SERVICES		TRAINING SERVICES		OTHER (SPECIFY)	

B. COMPANY INFORMATION:

PRIMARY PRODUCT(S) AT YOUR FACILITY? Industrial Chemicals
NUMBER OF EMPLOYEES AT YOUR FACILITY? 525
DO YOU HAVE AN ONGOING VIBRATION/PMP PROGRAM? YES ✓ NO ____
IF YES, HOW LONG? 7 years
PERSON RESPONSIBLE FOR PROGRAM? NAME: J.P. DenHartog
TITLE: Plant mgr.

C. WHAT TYPE OF VIBRATION INSTRUMENTS ARE USED AT YOUR PLANT? (ENTER BOTH MANUFACTURER MODEL NO. AND QUANTITY IN PARENTHESES)

- OVERALL LEVEL METER (i.e, SKF Picalog): SKF Picalog (3); CSI 2115 (4)
- ANALYZER/BALANCER (i.e., IRD 885): IRD 885(2)
- DATA COLLECTOR (i.e., IRD dataPAC 1500): CSI 2120(2)
- 1 CHANNEL RTA (i.e., ROCKLAND 5815A): ____
- MULTI-CHANNEL DATA COLLECTOR (i.e., CSI 2120): CSI 2400 (1)
- 2 CHANNEL RTA (i.e., HP 35670): ____
- SWEPT-FILTER ANALYZER (i.e., SCHENCK VIBRAPORT 30): IRD 880 (1)
- MULTI-CHANNEL RTA (i.e., SA 390): SA 390 (1)

D. WHAT IS THE BRAND NAME AND MODEL # OF THE PRIMARY DATA COLLECTOR USED AT YOUR PLANT? *(INDICATE MODEL NO. IN BLANK)*

B&K ____ CSI 2120 DI ____ DLI ____ ENTEK IRD ____
SKF ____ TEC ____ VITEC ____ OTHER ____

E. WHAT PMP SOFTWARE AND VERSION # DO YOU USE? (PLACE CHECK MARK BESIDE ALL THAT YOU NOW USE)

7090		DLI Advanced ALERT		EMONITOR (DOS)	
MASTERTREND (DOS)		MASTERTREND (WIN)	Ver 3.06	EMONITOR (WIN)	
IQ 2000		PRISM² (DOS)		OTHER	
		PRISM⁴ (WIN)			

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F. EXPERT DIAGNOSTIC SOFTWARE USED AT YOUR PLANT: (PLACE CHECK MARK)

AMETHYST	____	CHARLIE	____	EXAMINE	____	EXPLORE	____
EXPERTALERT	____	MDU	____	MOTORCHECK	____		
MOTORMONITOR	____	NSPECTR	✓	PRISM^4 PRO	____	ROMEX	____
VARMINT	____						

OTHER (PLEASE IDENTIFY) _____ NONE _____

G. EXPERIMENTAL MODAL ANALYSIS SOFTWARE USED AT YOUR PLANT (CHECK ALL THOSE WHICH YOU NOW USE).

LMS ____ MESCOPE ____ SDRC ____ STAR ✓ ZONIC ____ OTHER (SPECIF Y) ____

H. HAVE YOU PERFORMED FINITE ELEMENT MODELLING? IF SO, WITH WHAT SOFTWARE?

YES ____ NO ✓ SOFTWARE USED? _____

I. WHAT OTHER VIBRATION/PREDICTIVE MAINTENANCE SEMINARS HAVE YOU ATTENDED?

SEMINAR NAME	SEMINAR COMPANY	DATE ATTENDED (APPROXIMATE)
ENTRY	TECHNICAL ASSOC.	Jan. 1997

J. IS THERE ANY SPECIAL MACHINE TYPE IN YOUR FACILITY ON WHICH YOU NEED HELP ? AND/OR WOULD LIKE ADDITIONAL INFORMATION?

MACHINE NAME	BEARING TYPE	RPM	HP	DRIVE TYPE

K. PLEASE LIST ANY SPECIAL QUESTIONS OR TOPICS THAT YOU WOULD LIKE COVERED DURING THIS SEMINAR?

1) How can you accurately diagnose Rolling Element Brg problems and estmate their remaining life?

2) Is there any technique now available to detect problems on an intermediate gear inside a large multi-stage Gearbox?

L. HOW DID YOU FIND OUT ABOUT THIS SEMINAR? (PLEASE CHECK PRIMARY CONTACT)

BROCHURE	✓	DIRECT SALES	____	PHONE CALL	____
COLLEAGUE	____	FORMER ATTENDER	✓		

MAGAZINE AD (If So, Which Magazine)? _____

OTHER (PLEASE INDICATE SOURCE) _____

M. WHAT PROMPTED YOU TO ATTEND THIS PARTICULAR SEMINAR? (PLEASE CHECK ALL THAT APPLY.)

ADVANCEMENT	____	SEMINAR LOCATION	____	SPEAKER	____
SUBJECT MATTER	____	WAS RECOMMENDED	✓	OTHER	____

PLEASE SUBMIT THIS COMPLETED FORM AND MAIL/FAX TO TECHNICAL ASSOCIATES BEFORE ATTENDING SEMINAR. THANK YOU. WE APPRECIATE YOUR TAKING THE TIME TO HELP US CUSTOMIZE AND ENHANCE THIS SEMINAR TO ENSURE WE MEET YOUR PARTICULAR NEEDS.

TECHNICAL ASSOCIATES OF CHARLOTTE, P.C.

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TECHNICAL ASSOCIATES PRE-SEMINAR QUESTIONNAIRE

YOUR NAME: _____ SEMINAR DATES: _____
TITLE: _____ SEMINAR LOCATION: _____
COMPANY: _____ SEMINAR LEVEL (CIRCLE) ENTRY I II III ADV
LOCATION: _____ VIBRATION EXPERIENCE: YRS. ____ MOS. ____
PHONE: _____ EXT: _____ SPECTRAL ANAL. EXPER: YRS. ____ MOS. ____
EMAIL: _____ FAX: _____

PLEASE COMPLETE THE FOLLOWING QUESTIONNAIRE, THIS INFORMATION IS FOR USE IN THIS CLASS AND TO HELP US PLAN FUTURE SEMINARS: ***(SEE SAMPLE PRE-SEMINAR SURVEY ENCLOSED)***

A. TYPE OF INDUSTRY AND PRODUCT MANUFACTURED/SERVICES RENDERED BY YOUR COMPANY:

AUTOMOTIVE	_____	CHEMICAL	_____	CONSUMER PROD.	_____
CONSULTANT	_____	FOOD/BEVERAGES	_____	FABRICA. METAL	_____
FIBERGLASS	_____	HVAC	_____	INSTRUM./S'WARE	_____
MINING	_____	PETROLEUM	_____	PHARMACEUTICAL	_____
PULP & PAPER	_____	REPAIR	_____	SYNTHETIC FIBER	_____
TEXTILES	_____	TOBACCO	_____	UTILITIES	_____
VIB. SERVICES	_____	TRAINING SERVICES	_____	OTHER (SPECIFY)	_____

B. COMPANY INFORMATION:

PRIMARY PRODUCT(S) AT YOUR FACILITY? _____
NUMBER OF EMPLOYEES AT YOUR FACILITY? _____
DO YOU HAVE AN ONGOING VIBRATION/PMP PROGRAM? YES _____ NO _____
IF YES, HOW LONG? _____
PERSON RESPONSIBLE FOR PROGRAM? NAME: _____
TITLE: _____

C. WHAT TYPE OF VIBRATION INSTRUMENTS ARE USED AT YOUR PLANT? (ENTER BOTH MANUFACTURER MODEL NO. AND QUANTITY IN PARENTHESES)

- OVERALL LEVEL METER (i.e, SKF Picalog) _____
- DATA COLLECTOR (i.e., ENTEK DataPAC 1500) _____
- MULTI-CHANNEL DATA COLLECTOR (i.e., CSI 2120) _____
- SWEPT-FILTER ANALYZER (i.e., SCHENCK VIBRAPORT 30) _____
- ANALYZER/BALANCER (i.e., IRD 885) _____
- 1 CHANNEL RTA (i.e., ROCKLAND 5815A) _____
- 2 CHANNEL RTA (i.e., HP 35670) _____
- MULTI-CHANNEL RTA (i.e., SA 390) _____

D. WHAT IS THE BRAND NAME AND MODEL # OF THE PRIMARY DATA COLLECTOR USED AT YOUR PLANT? *(INDICATE MODEL NO. IN BLANK)*

B&K _____ COMMTEST _____ CSI _____ DI _____ DLI _____
ENTEK IRD _____ SKF _____ TEC _____ VITEC _____ OTHER _____

E. WHAT PMP SOFTWARE AND VERSION # DO YOU USE? (PLACE CHECK MARK BESIDE ALL THAT YOU NOW USE)

ALERT _____	EMONITOR (DOS) _____	IQ 2000 _____
MAS _____	MASTERTREND _____	ODYSSEY _____
PRISM^4 _____	RBMWARE _____	OTHER _____

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F. EXPERT DIAGNOSTIC SOFTWARE USED AT YOUR PLANT: (PLACE CHECK MARK)

AMETHYST _____ CHARLIE _____ EXAMINE _____ EXPLORE _____
EXPERTALERT _____ MDU _____ MOTORCHECK _____
MOTORMONITOR _____ NSPECTR _____ PRISM^4 PRO _____ ROMEX _____
VARMINT _____
OTHER (PLEASE IDENTIFY) _____ NONE _____

G. EXPERIMENTAL MODAL ANALYSIS SOFTWARE USED AT YOUR PLANT (CHECK ALL THOSE WHICH YOU NOW USE).

LMS ____ MESCOPE ____ SDRC ____ STAR ____ ZONIC ____ OTHER (SPECIFY) ____

H. HAVE YOU PERFORMED FINITE ELEMENT MODELLING? IF SO, WITH WHAT SOFTWARE?

YES_____ NO_____ SOFTWARE USED?_____

I. WHAT OTHER VIBRATION/PREDICTIVE MAINTENANCE SEMINARS HAVE YOU ATTENDED?

SEMINAR NAME	SEMINAR COMPANY	DATE ATTENDED (APPROXIMATE)

J. IS THERE ANY SPECIAL MACHINE TYPE IN YOUR FACILITY ON WHICH YOU NEED HELP ? AND/OR WOULD LIKE ADDITIONAL INFORMATION?

MACHINE NAME	BEARING TYPE	RPM	HP	DRIVE TYPE

K. PLEASE LIST ANY SPECIAL QUESTIONS OR TOPICS THAT YOU WOULD LIKE COVERED DURING THIS SEMINAR?

_____
_____
_____
_____
_____
_____
_____

L. HOW DID YOU FIND OUT ABOUT THIS SEMINAR? (PLEASE CHECK PRIMARY CONTACT)

BROCHURE _____ DIRECT SALES _____ PHONE CALL _____
COLLEAGUE _____ FORMER ATTENDER _____
MAGAZINE AD (If So, Which Magazine)? _____
OTHER (PLEASE INDICATE SOURCE)_____

M. WHAT PROMPTED YOU TO ATTEND THIS PARTICULAR SEMINAR? (PLEASE CHECK ALL THAT APPLY.)

ADVANCEMENT _____ SEMINAR LOCATION _____ SPEAKER _____
SUBJECT MATTER _____ WAS RECOMMENDED _____ OTHER _____

PLEASE SUBMIT THIS COMPLETED FORM AND MAIL/FAX TO TECHNICAL ASSOCIATES BEFORE ATTENDING SEMINAR. THANK YOU. WE APPRECIATE YOUR TAKING THE TIME TO HELP US CUSTOMIZE AND ENHANCE THIS SEMINAR TO ENSURE WE MEET YOUR PARTICULAR NEEDS.

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POST-SEMINAR EVALUATION

SEMINAR DATES: _____ SEMINAR LEVEL (CIRCLE): ENTRY I II III ADV.

ATTENDEE NAME: _____ TITLE: _____

COMPANY NAME: _____ YOUR VIBRATION EXPERIENCE: YEARS _____ MONTHS _____

COMPANY LOCATION: _____ YOUR SPECTRAL ANAL. EXPER: YEARS _____ MONTHS _____

SEMINAR LOCATION: _____

INSTRUCTOR: _____

EVALUATION TOPICS	LOW									HIGH
RATING >	1	2	3	4	5	6	7	8	9	10
1. EVALUATE CONTENT OF THIS SEMINAR.										
2. EVALUATE PERFORMANCE OF INSTRUCTOR:										
A. KNOWLEDGE OF SUBJECT										
B. PRESENTATION CAPABILITIES										
C. ENTHUSIASM DISPLAYED										
D. OVERALL PERFORMANCE RATING										
3. HOW APPLICABLE TO YOUR JOB HAS BEEN THE CONTENT OF THIS SEMINAR AND SEMINAR TEXT?										
4. CAN WHAT YOU HAVE LEARNED MAKE YOU MORE EFFECTIVE IN YOUR JOB?										
5. HOW WOULD YOU RATE THE SEMINAR FACILITIES:										
A. MEETING ROOM -										
B. SLEEPING ROOM -										
C. FOOD SERVICE -										
6. ARE YOU INTERESTED IN FURTHER TRAINING IN THIS AREA OF VIBRATION/PREDICTIVE MAINTENANCE?										

CONCENTRATION ON SEMINAR:

1. ON WHAT SHOULD ***MORE*** TIME BE CONCENTRATED? _____

2. ON WHAT SHOULD ***LESS*** TIME BE CONCENTRATED? _____

3. WHAT WOULD YOU SUGGEST TO ***IMPROVE*** FUTURE SEMINARS? _____

4. DO YOU HAVE ANY ***ADDITIONAL COMMENTS*** ABOUT THIS SEMINAR, THE INSTRUCTORS, THE COMPANY OR THE FACILITIES? _____

WE WANT TO GENUINELY THANK YOU FOR THE PRIVILEGE OF SHARING WITH YOU THIS WEEK. WE HOPE TO SEE YOU AGAIN IN THE FUTURE.

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RECOMMENDED PERIODICALS FOR THOSE INTERESTED IN PREDICTIVE MAINTENANCE

1. *Sound and Vibration Magazine*

P.O. Box 40416
Bay Village, OH 44140
Mr. Jack Mowry, Editor and Publisher
Phone: 216-835-0101
Fax : 216-835-9303

Terms: Normally free for bona fide qualified personnel concentrating in the Sound and Vibration Analysis/Plant Engineering Technologies. Non-qualified personnel - $25/per year within the U.S.

Comments: This is a monthly publication that normally will include approximately 4-6 issues per year devoted to Predictive Maintenance. Their Predictive Maintenance articles are usually practical and in good depth; normally contain real "meat" for the PPM vibration analyst. *Sound and Vibration* has been published for over 25 years.

2. *Vibrations Magazine*

The Vibration Institute
6262 South Kingery Hwy, Suite 212
Willowbrook, IL 60514
Institute Director - Dr. Ronald Eshleman
Phone: 630-654-2254
Fax : 630-654-2271

Terms: *Vibrations Magazine* is sent to Vibration Institute members as part of their annual fee, (approx. $45 per year). It is available for subscription to non-members at $55/per year; $60/foreign.

This is a quarterly publication of the Vibration Institute. Always contains very practical and useful Predictive Maintenance Articles and Case Histories. Well worth the small investment.

Comments: Yearly Vibration Institute fee includes reduced proceedings for that year if desired for the National Conference normally held in June. They normally meet once per year at a fee of about $675/per person, ($600/person for Institute members) including conference proceedings notes and mini-seminar papers. All of the papers presented, as well as mini-courses, at the meeting are filled with "meat" for the Predictive Maintenance Vibration Analyst. *Vibrations Magazine* was first published in 1985 although the Institute has been in existence since approximately 1972, with their first annual meeting in 1977. The Vibration Institute has several chapters located around the United States which normally meet on a quarterly basis. The Carolinas' Vibration Institute Chapter normally meets in Greenville, SC; Charleston, SC; Columbia, SC; Charlotte, NC; Raleigh, NC; and in the Winston Salem, NC areas. For Institute membership information, please contact: Dr. Ron Eshleman at 630-654-2254. When doing so, be sure to ask what regional chapter is located to your area. Membership fees for the "Annual Meeting Proceedings" are $30/per year (normal cost is approx. $60/per year for proceedings if annual meeting is not attended). Please tell Ron that we recommended you joining the Vibration Institute when you call or write to him.

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3. *P/PM Technology Magazine*

P.O. Box 1706
Minden, NV 89423-1706 (Pacific Coast Time)
Phone: 702-267-3970; 800-848-8324
Fax : 702-267-3941
Publisher- Mr. Ronald James; Assistant Publisher: Susan Estes

Terms: $42/per year for qualified USA subscribers, (individuals and establishments involved with industrial plant and facilities maintenance; subscribers must be associated in engineering, maintenance, purchasing or management capacity). $60/year for unqualified subscribers.

Comments: This is a bi-monthly magazine with articles about all facets of PPM Technologies, including Vibration Analysis, Oil Analysis, Infrared Thermography, Ultrasonics, Steam Trap Monitoring, Motor Current Signature Analysis, etc. These are normally good practical articles. Also includes some cost savings information, although does not necessarily include how these cost savings were truly determined. *P/PM Technology* also hosts at least one major conference per year in various parts of the United States. Intensive training courses in a variety of condition monitoring technologies will also be offered in vibration analysis, root cause failure analysis, oil analysis, thermographic analysis, ultrasonic analysis, etc..)

4. *Maintenance Technology Magazine*

1209 Dundee Ave., Suite 8
Elgin, IL 60120
Phone: 800-554-7470
Fax : 804-304-8603
Publisher: Arthur L. Rice

Terms: $95/per year for non-qualified people This is a monthly magazine that usually has at least one article relating to Predictive Maintenance using vibration analysis within each issue. In addition to vibration, it likewise always offers other articles covering the many other technologies now within Predictive Maintenance.

5. Reliability Magazine

PO Box 856
Monteagle, TN 37356
Phone: 423-592-4848
Fax : 423-592-4849

Editor: Mr. Joseph L. Petersen

Terms: $49 per year in USA; $73 per year outside USA.

Comments: This bi-monthly magazine covers a wide variety of Condition Monitoring Technologies including Vibration Analysis, Training, Alignment, Infrared Thermography, Balancing, Lubrication Testing, CMMS and a unique category they entitle "Management Focus".

NOTE: In addition to these periodicals, many of the major predictive maintenance hardware and software vendors put out periodic newsletters. Some of these in fact do include some “real meat” in addition to their sales propaganda. We would recommend that you contact, particularly the vendor supplying your predictive maintenance system for their newsletter. Their newsletter will likewise advise you of updates in their current products.

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CHAPTER 1

ANALYSIS II SEMINAR OVERVIEW

An effective Predictive Maintenance Program (PMP) is a total program of:

1. ***DETECTION***
2. ***ANALYSIS***
3. ***CORRECTION***
4. ***VERIFICATION***

Therefore, these 4 important steps will formulate the guiding philosophy which will form the foundation for this "ANALYSIS II" seminar. Our goal will be to provide the tools an analyst needs to detect the very onset of problems within machinery included in his program. It will then provide extensive diagnostic techniques required to analyze machine problems to determine both their cause and severity. It will then put one in a position to make solid recommendations based on fact rather than "feeling", and will allow the maintenance department to schedule such corrective measures at convenient times. Finally, this seminar will provide instruction on how to verify that corrective measures did in fact correct the problem(s), and that no new problems have been introduced.

Following the completion of this course, the student should have a solid working knowledge of the proper application instrumentation and software required for both setting up and implementing an effective condition monitoring program, as well as significantly enhance his knowledge on how to effectively troubleshoot mechanical and electrical problems within machinery using vibration analysis and related nondestructive technologies. Following below are brief introductions for each of the chapters which will be covered in this seminar text:

- **CHAPTER 2 - "BRIEF REVIEW OF 'ANALYSIS I' SEMINAR TOPICS":** This chapter will review some of the more important topics which are covered in the "ANALYSIS I" seminar. Initially, it will review the fundamentals of vibration analysis and how it can be used to evaluate machine condition. Next, it will provide a generic overview of the various types of vibration instruments available today, including both their strengths and their weaknesses. This includes a comprehensive table which summarizes a great number of capabilities which are possible with vibration analysis, and clearly identifies which of the instrument types (not specific vendors) can perform which of the tabulated tasks. Finally, the chapter closes by reviewing each of the major vibration transducers available today and gives important instruction on how to properly select the right transducer for the particular job or test to be performed.

- **CHAPTER 3 - "PRINCIPLES OF DIGITAL DATA ACQUISITION AND FFT PROCESSING FOR RELIABLE SPECTRAL ANALYSIS":** This comprehensive chapter provides the analyst with important information on how his vibration analyzer acquires and processes the data which is so critical to the success of his program. Often, seminars simply assume the analyst understands these principles and never provide any real background on just how the instruments function, what effects it might make on their data if they do not understand this information, and how they might best optimize their instrument and supporting software to acquire the data they need to properly evaluate the machines or structures in question. More specifically, this chapter will examine FFT properties, sampling and digitizing of data, aliasing, window selection, types of averaging, overlap processing, the importance of bandwidth in ensuring all frequencies from all sources are displayed, the effect of dynamic range on frequency and amplitude display, and a discussion on the distinct difference between digital and analog acquisition of the overall vibration level.

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- **CHAPTER 4 - "INTRODUCTION TO NATURAL FREQUENCY TESTING & INSTRUMENTATION":** This chapter will introduce the analyst to the difference between the terms "natural frequency", "resonance" and "critical speed". It will then provide invaluable information on how to perform a variety of natural frequency tests, including both the instrumentation and accessories required to accurately perform these tests. Test methods taught will include impulse natural frequency tests, coastdown and runup tests; and how to both acquire and interpret Bode' and polar plots which confirm the presence of natural frequencies (some of which may be found to be resonances of the machinery being evaluated). In these cases, introductory information will be provided on how to go about correcting such resonance problems.

- **CHAPTER 5- "ENHANCED VIBRATION DIAGNOSTICS USING PHASE ANALYSIS AND CASCADE DIAGRAMS":** Describes the various instruments and transducers required to measure phase as well as those to generate cascade diagrams (sometimes called "waterfall plots" or "spectral maps"). This chapter points out that phase is the "third leg of the triangle" which describes machine and structural vibration response. That is, these "three legs of the triangle" include amplitude (how much vibration), frequency (how many cycles of vibration per unit of time) and phase (which describes the vibration at one location relative to the vibrating motion at another location). This chapter points out that by taking phase measurements in the horizontal, vertical and axial directions on each bearing housing, one can determine whether a problem showing high vibration at 1X RPM is unbalance, misalignment, soft foot, bent shaft, eccentric rotor, loose hold-down bolts, resonance, cocked bearing, a combination of these problems, or several other potential problems (all of which can generate vibration spectra which "appear" to be identical). This chapter then takes a close look at the value and optimum utilization of cascade diagrams pointing out how they give the analyst a unique view of how the vibration response changes either over a short period of time (for example, during either a runup or coastdown), or over a rather long period of time (for example, from one month to the next during PMP surveys). Instruction is provided on how to use this information to help diagnose a number of problems including rotor rub, resonance, oil whirl, oil whip, etc.

- **CHAPTER 6 - "CONCENTRATED VIBRATION SIGNATURE ANALYSIS TO DETECT A SERIES OF MECHANICAL AND ELECTRICAL PROBLEMS":** Chapter 6 probably forms the "centerpiece" of this seminar and has been widely acclaimed by attendees over the years as the key which has helped them significantly elevate the effectiveness of their programs. This chapter will introduce Technical Associates' world renown **"Illustrated Vibration Diagnostic Wall Chart"** which will review theory on how to detect some 44 machine problems, including those from both mechanical and electrical problem sources. This chapter begins with a review of the less complex problems which are covered in the "ANALYSIS I" seminar; and then provides in-depth instruction on how to detect rotor rub, journal bearing, gear, electrical, beat vibration, resonance and rolling element bearing problems (including a series of "failure scenarios" which have been identified through the years to track the condition of rolling element bearings).

- **CHAPTER 7 - "PROVEN METHOD FOR SPECIFYING BOTH SIX SPECTRAL ALARM BANDS AS WELL AS NARROWBAND ALARM ENVELOPES USING TODAY'S PREDICTIVE MAINTENANCE SOFTWARE SYSTEMS":** After covering how to detect the whole series of mechanical and electrical problems, as well as revealing where they will appear in a vibration spectrum in Chapter 6, Chapter 7 next shows one how to properly specify both spectral band as well as narrowband envelope alarms, not only to detect the presence of such problems, but also to give the analyst plenty of time to react and take the required corrective measures before potential catastrophic failures. These documented methods (which were developed over the past 15 years by implementing a series of predictive maintenance programs on a broad range of machinery) have received wide acclaim and, like the signature analysis theory taught in Chapter 6, have often received much credit for greatly enhancing the effectiveness of condition monitoring programs by allowing the analysts to concentrate their efforts on the machines truly in need of attention.

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- **CHAPTER 8 - "INTRODUCTION TO LISSAJOUS ORBIT ANALYSIS":** Introduces the analyst to Lissajous orbit plots which show the actual path the shaft itself follows inside the bearing. It points out that Lissajous patterns can be used to study the shaft dynamic behavior, measure the relative phase angle between motions at different points on the structure, and to detect the presence of machine faults such as mechanical runout, eccentricity, misalignment, rotor rub, gear and bearing faults. This chapter provides this information in practical terms and provides real-world examples of how Lissajous pattern recognition has been so successful in detecting numerous problems which might otherwise have gone undetected had this technique not been employed.

- **CHAPTER 9 - "INTRODUCTION TO HIGH FREQUENCY DEMODULATED AND ENVELOPED SPECTRA":** Describes how these tools which are now available on many of today's programmable data collectors can be used to provide an early warning of impending problems with rolling element bearings, gears, cavitation, lubrication, electrical faults, etc. Although these techniques have been around for some years, only recently since approximately 1990 have they began to appear within many of today's data collectors. The problem is that so few analysts yet understand what high frequency enveloping technology is, much less how it works. Therefore, the expressed purpose of this chapter is to initially provide the analyst with a fundamental understanding of how this data is acquired, how it is processed and how to interpret the resulting spectra once they have been generated. Then, if this tool is used along with vibration spectral analysis, they provide a powerful set of diagnostic tools which can not only detect problems at early stages, but also can track their deterioration and allow their correction before the damage to the machine is extensive.

- **CHAPTER 10 - "INTRODUCTION TO VIBRATION ISOLATION MECHANISMS":** Introduces the analyst to a variety of vibration isolation mechanisms and points out the distinct difference between the terms "vibration isolation" and "vibration damping" which unfortunately are often used by many to mean the same thing. This chapter, along with Chapter 11, points out the distinct difference between the two terms and describes how each of these two vibration treatment methods function on a very practical level. This chapter not only discusses the theory and provides illustrated examples of some of the more popular isolators available, but also gives good "rules of thumb" on how to specify proper isolation treatments (in order to avoid amplifying vibration rather than isolating it). Likewise, it includes an invaluable table comparing many of the isolator types and showing what frequencies they will and will not isolate. This chapter also includes a real-world example of how isolation was employed to prevent vibration originating from a two-stage reciprocating air compressor from transmitting into an electron microscope directly above it on the second floor even though the compressor itself was installed only 21 inches away from a load bearing wall common to the building structure of both the compressor room and microscope lab.

- **CHAPTER 11 - "INTRODUCTION TO VIBRATION DAMPING TREATMENTS":** Introduces the analyst to the theory of vibration damping on a practical level. This chapter points out this parameter (damping) is probably the most misunderstood of any of the three major parameters effecting vibration response of a machine or structure (stiffness, mass and damping). It provides a good definition and description of the theory of damping in everyday, practical terms rather than the usual, highly technical jargon normally associated with discussions on this topic. Likewise, it also provides information on some of the more popular damping treatments; and, importantly, points out when damping treatments should be used, as well as when installation of damping materials is likely to be a waste of time and funding.

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- **CHAPTER 12 - "REAL-WORLD CASE HISTORIES OF VIBRATION DIAGNOSTICS CONDUCTED ON VARIOUS MACHINE TYPES":** Offers an array of actual case histories which have been performed in order to give the student a taste of how such problems were solved on actual machines using the tools taught in the seminar. This section includes over 250 pages of such case histories showing how such problems as rotor rub, gear, electrical, resonance and rolling element bearing wear were detected, and subsequently corrected without catastrophic failure. Impressive "before" and "after" frequency spectra are included to show the effect of properly taking the recommended corrective actions on the machine, and thereby prolonging the life of the equipment by reducing these vibration amplitudes.

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CHAPTER 2

BRIEF REVIEW OF "ANALYSIS I" SEMINAR TOPICS

2.0 INTRODUCTION

Included in this section will be a brief review of some of the topics which were covered in the "ANALYSIS I" seminar. This particular chapter will review these particular topics:

2.1 WHAT IS VIBRATION AND HOW CAN IT BE USED TO EVALUATE MACHINERY CONDITION?

2.2 OVERVIEW OF THE STRENGTHS AND WEAKNESSES OF TYPICAL VIBRATION INSTRUMENTS

2.3 OVERVIEW OF VIBRATION TRANSDUCERS AND HOW TO PROPERLY SELECT THEM

2.4 UNDERSTANDING VIBRATION PHASE AND ITS APPLICATIONS

In addition to these topics, other items which were covered in "ANALYSIS I" will be briefly reviewed in other sections of the "ANALYSIS II" seminar text. However, the expressed purpose of Chapter 2 is to ensure everyone reviews the fundamentals before proceeding to more advanced topics.

2.1 WHAT IS VIBRATION AND HOW CAN IT BE USED TO EVALUATE MACHINERY CONDITION?

2.11 INTRODUCTION

Vibration is the response of a system to an internal or external stimulus causing it to oscillate or pulsate. While it is commonly thought that vibration itself damages machines and structures, it does not. Instead, *the damage is done by dynamic stress which causes fatigue of the materials;* and the *dynamic stresses are induced by vibration*. Equation 1 shows that the Vibration Amplitude is directly proportional to the Dynamic Force, and inversely proportional to the Dynamic Resistance in a spring-mass system like that shown in Figure 1. That is, *if two machines are subject to the same dynamic force, the amplitude response from the machine which has greater dynamic resistance will be less* than that of the other machine. For example, if a machine is placed on spring isolators, the vibration will likely increase due to less dynamic resistance for the same imposed dynamic forces. The transmission of vibration to the floor and surrounding structures will be less, but the vibration within the machine will likely increase. Yet, no additional damage will be done to the machine since the same *forces* (and therefore, fatigue stresses) will remain the same within this machine (as compared to when the machine was directly mounted to

$$\text{VIBRATION AMPLITUDE RESPONSE} \propto \frac{\text{DYNAMIC FORCE}}{\text{DYNAMIC RESISTANCE}} \qquad \textbf{Eqn. 1}$$

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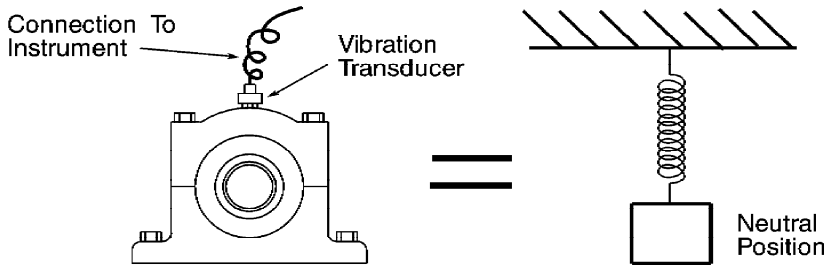

FIGURE 1
MASS IN NEUTRAL POSITION WITH NO APPLIED FORCE

the floor). *Dynamic Resistance within a machine or structure is proportional to the amount of stiffness, damping and mass within the system.* This will be discussed later in Chapter 11 which examines how these 3 parameters interact with one another.

Vibration has three important parameters which can be measured:

1. **Frequency** - How many times does the machine or structure vibrate per minute or per second?

2. **Amplitude** - How much vibration in mils, in/sec or g's?

3. **Phase** - How is the member vibrating in relation with a reference point?

2.12 WHAT IS VIBRATION FREQUENCY AND HOW DOES IT RELATE TO A TIME WAVEFORM?

Recall from an example of a pencil trace drawn on a strip chart recorder (if the pencil was fastened to a suspended mass which oscillates up and down on a spring), a uniform series of sine waves would be drawn. *Each sine wave would represent one completed cycle* - the mass would go from its neutral position to an upper limit of travel, down through its neutral position, then down to a lower limit of travel, and finally back to its neutral position (this completes one cycle of motion). *Figure 2 shows how frequency can be calculated from it by measuring the time period (T) of one cycle (sec/cycle) and inverting to determine the frequency (cycles/sec). This is an example of a time waveform which plots Vibration Amplitude versus Time*. This waveform is a truly sinusoidal waveform from which direct comparisons can be made between its Peak-to-Peak, Peak and RMS amplitudes (this will be covered in another section).

Frequency is expressed in either Cycles per Minute (CPM) or in Cycles per Second (CPS), which is now called Hertz (where 1 Hertz or Hz = 60 CPM).

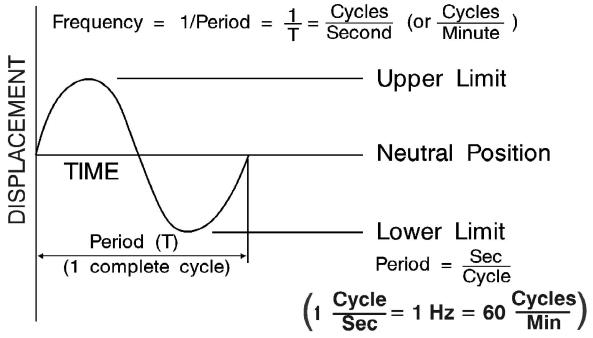

FIGURE 2
DISPLACEMENT AND FREQUENCY FROM A TIME WAVEFORM

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When is a good time to use time waveforms in an analysis? Time waveforms are an excellent analytical tool to use when analyzing gearboxes. The transducer can be attached close to the input or the output shaft bearing to check for broken or chipped gear teeth. The following is a typical example of how a display for one broken tooth would appear as a time waveform, shown in Figure 3.

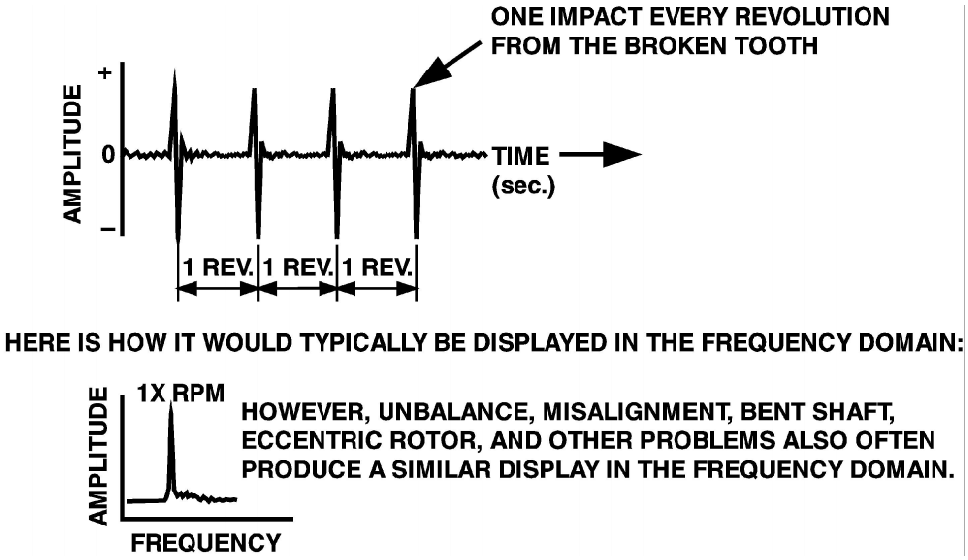

FIGURE 3
HOW A BROKEN TOOTH ON A GEAR IS DISPLAYED IN TIME WAVEFORM AND IN A SPECTRUM

Time waveforms are especially ideal for low-speed shafts and gears, even if some never rotate a full revolution (basically just rocking back and forth). In this case, time waveforms are virtually the only analytical tool which can be effectively used.

In the time waveform shown in the above example, an analyst can calculate the frequency of the impact or the speed of the shaft even though the display is in the time domain. If the time between each impact was given as 5 milliseconds (.005 second), the frequency would be calculated as:

$$F = \frac{1}{T}$$
$$= \frac{1}{.005 \text{ SECOND/CYCLE}}$$
$$= 200 \text{ CYCLES/SECOND}$$
$$= 200 \text{ Hz}$$
$$= 200 \text{ CYC/SEC X } 60 \text{ SEC/MIN}$$
$$= 12{,}000 \text{ CPM}$$

Thus, the frequency of the impacts (or the speed of the shaft in this case) is 12,000 CPM. Likewise, it can be readily seen that if the time between impacts was 5 seconds instead, the frequency would only be .20 Hz (1/5 = .20 cyc/sec) or 12 CPM - a very low frequency indeed. All this can be determined from a time waveform.

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2.13 WHAT IS VIBRATION AMPLITUDE?

2.131 What is Vibration Displacement?

Displacement is a measure of the *total travel* of the mass - back and forth. Displacement can be expressed in mils (where 1 mil = .001 inch, or in microns (where 1 micron, m = .001 millimeter or .039 mil). When a machine is being subjected to excessive dynamic stress at very low frequencies, displacement may be a good indicator of vibration severity since the machine (or structure) may be flexing too much; or simply being bent too far.

2.132 What is Vibration Velocity?

The velocity of the vibration is *a measure of the speed* at which the mass is moving or vibrating during its oscillations. The faster a machine flexes, the sooner it will fail in fatigue. ***Vibration velocity is directly related to fatigue.*** Note from the example of the oscillating mass suspended from a spring in Figure 4, that *velocity reaches its maximum value* (or peak) *at the neutral position where the mass is fully accelerated (acceleration is zero) and now begins to decelerate* as shown in Figure 4. Velocity is expressed as inches per second (in/sec) or as millimeters per second (mm/sec).

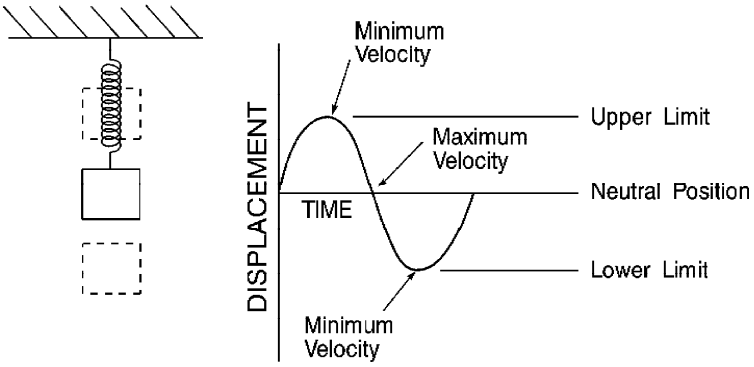

FIGURE 4
VELOCITY FROM THE DISPLACEMENT CURVE

However, if an analyzer was used to directly measure peak velocity, it would select the highest peak or excursion that the velocity time waveform would make. From an oscilloscope display, the peak velocity would be the highest peak in the display as shown in Figure 5.

FIGURE 5
HOW TO DETERMINE PEAK VELOCITY FROM AN OSCILLOSCOPE DISPLAY

In this case, the peak velocity is .7 in/sec because it is the highest peak, positive *or* negative.

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2.133 What is Vibration Acceleration?

When a machine housing vibrates, it experiences acceleration since it continually changes speed as it oscillates back and forth. *Acceleration is greatest at the instant at which velocity is at its minimum. That is, this is the point where the mass has decelerated to a stop and is about to begin accelerating (moving faster) in the opposite direction*. Acceleration is the rate of change in velocity and is measured in units of g's (where 1g = 32.2 ft/sec^2 = 386 in/sec^2 = 22.0 mi/hr per second change). *The greater the rate of change of velocity, the higher will be the forces (and stresses) on this machine due to the higher rate of acceler*ation. *At high frequencies*, failure of a machine may result from *excessive forces* which break down the lubrication allowing surface failures of bearings (due to metal-to-metal contact). These excessive forces are directly proportional to acceleration (F=ma). Acceleration is probably the most difficult measure of vibration amplitude to grasp, but is the parameter most often directly measured in the field with the use of an accelerometer. Thus, it is important that an analyst gain a good understanding of it.

2.14 WHAT IS VIBRATION PHASE?

Phase is a measure of how one part is moving (vibrating) in relation to another part, or to a fixed reference point. Vibration phase is measured in angular degrees by using either a strobe light or an electronic photocell. Figure 6 shows two masses vibrating with a 90° phase difference. That is, Mass #2 is one-fourth of a cycle (or 90°) ahead of Mass #1; thus, *Mass #2 is "leading" Mass #1 in phase by 90°*. Or, from the other point of view, *Mass #1 has a 90° phase lag relative to the motion of Mass #2*.

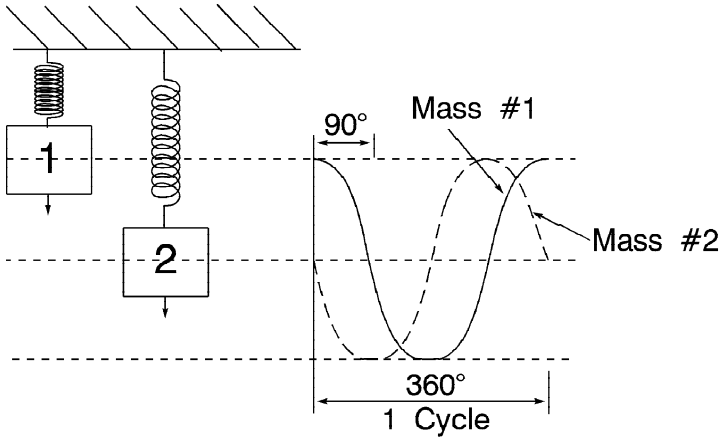

FIGURE 6
TWO MASSES WITH 90° PHASE DIFFERENCE

Figure 7 shows the same two masses vibrating with an *180° phase difference*. That is, at any instant in time, Mass #1 will move downwards at the same instant as Mass #2 moves upwards, and vice versa.

Figure 8 shows how phase relates to machine vibration. The left sketch shows a 0° phase difference between bearing Positions 1 and 2 (in-phase motion). The right sketch shows a 180° out-of-phase difference between these positions (out-of-phase motion).

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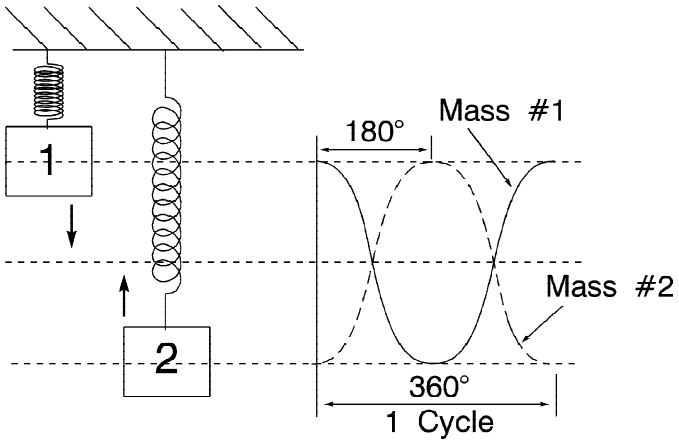

FIGURE 7
TWO MASSES WITH 180° PHASE DIFFERENCE

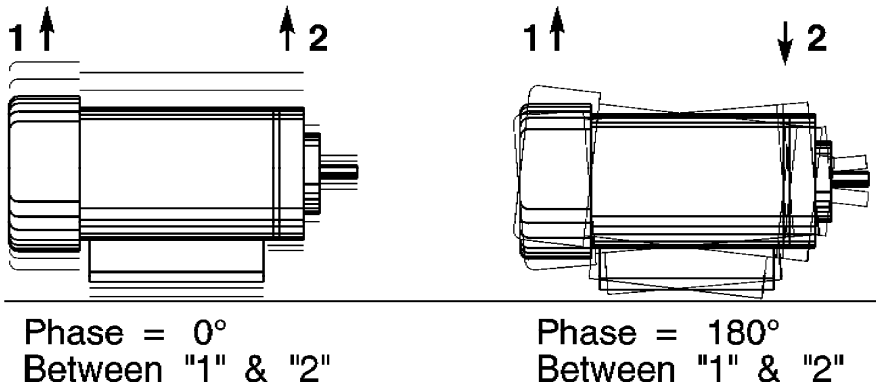

FIGURE 8
PHASE RELATIONSHIP AS USED WITH MACHINERY VIBRATION

2.141 How To Read Phase on CRT or RTA Screens

a. At the dashed lines, the following illustrations of various time waveforms show that the same position on each wave maintains the same phase relationship. That is, for 90° or any other angle on each wave, the 90° location (or any other angle) remains the same on all waves regardless of how the waveform is displayed (that is, the location of 90° is at the highest positive point; 180° is at zero amplitude with the waveform sloping downwards; 270° is at the lowest negative point; while 0° (or 360°) is back at zero amplitude, but with the waveform sloping upwards (positive slope).

Another point to be made about the waveforms of Figure 9 is to show how waveforms can be used to compare phase at various locations. For example, Waveform A might be at the driver outboard bearing horizontal; Waveform B at the driver inboard bearing horizontal; while Waveform C is at the inboard bearing of the driven machine. *If all 3 waveforms were captured simultaneously, phase comparisons can be made*. In this case, it would show there is a 180° phase difference between Waveforms A and B (when A goes up, B goes down and vice versa). On the other hand, there is only a 90° phase difference between Waveforms A and C.

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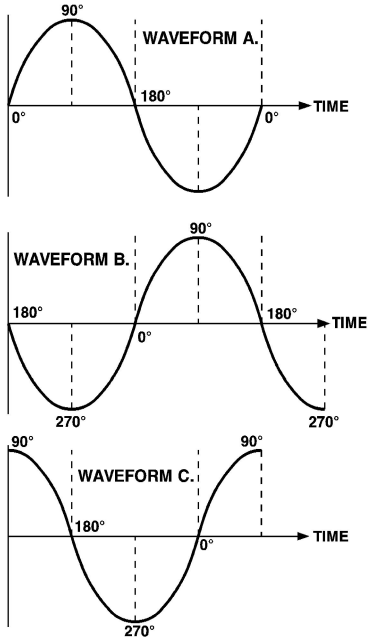
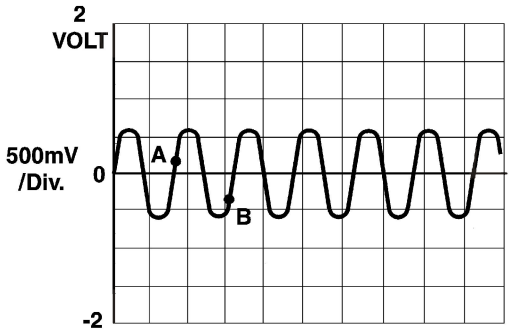


FIGURE 9
HOW TO DETERMINE THE PHASE DIFFERENCE BETWEEN TWO TIME WAVEFORMS

b. How to determine the phase difference between two points on the same time waveform:



PROBLEM: WHAT IS THE PHASE DIFFERENCE (SHAFT) BETWEEN POINT A AND POINT B ON THE DISPLAY, IN DEGREES ?

SOLUTION 1

1) PHASE FOR POINT A = 30°
2) PHASE FOR POINT B = 320°
3) THE DIFFERENCE IS:
FROM 30° → 90° = 60°
FROM 90° → 270° = 180°
FROM 270° → 320° = 50°
ADD 290°

SOLUTION 2

1) COUNT THE NUMBER OF PEAKS ACROSS THE 10 DIVISIONS.
THERE ARE APPROXIMATELY 6.4 PEAKS.
THERE ARE APPROXIMATELY 6.4 CYCLES.

2) DIVIDE 6.4 CYCLES INTO 10 DIVISIONS.
THERE ARE $\frac{10}{6.4} = 1.56 \frac{\text{DIVISIONS}}{\text{CYCLES}}$

3) MEASURE THE DISTANCE IN DIVISIONS BETWEEN POINT A POINT B.
THERE ARE APPROXIMATELY 1.25 DIVISIONS.

4) THERE ARE 360° IN ONE CYCLE,
THERE ARE $\frac{1.25}{1.56} \times 360° = 288° (290°)$
DIFFERENCE BETWEEN POINTS A & B.

FIGURE 10
HOW TO DETERMINE THE PHASE DIFFERENCE BETWEEN TWO POINTS ON THE SAME SINUSOIDAL TIME WAVEFORM

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2.142 Phase Relationship of Acceleration, Velocity & Displacement Time Waveforms

Figure 11 shows the phase relationship between acceleration, velocity and displacement time waveforms. *It shows that acceleration leads velocity by 90° and leads displacement by 180°.* On the other hand velocity *lags* acceleration by 90°, but *leads* displacement by 90°. Finally displacement *lags* acceleration by 180° and *lags* velocity by 90°.

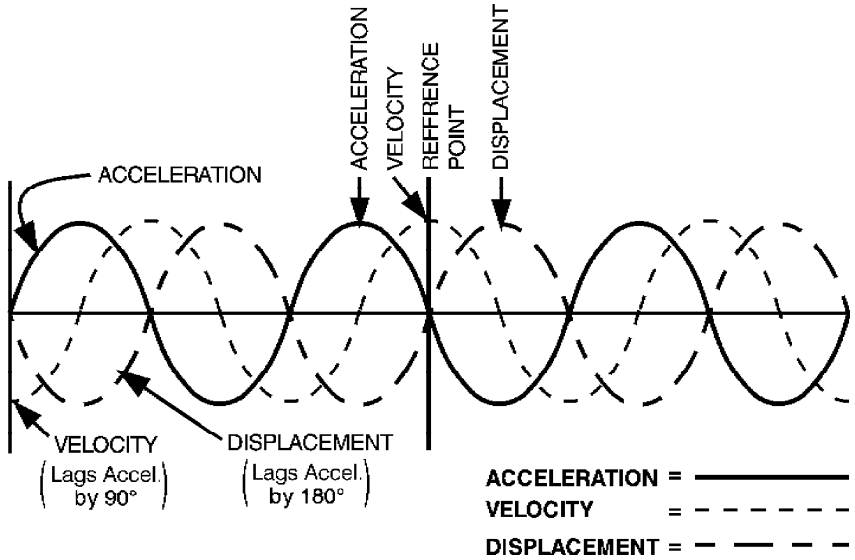

FIGURE 11
PHASE RELATIONSHIP BETWEEN ACCELERATION, VELOCITY AND DISPLACEMENT TIME WAVEFORMS

2.15 WHAT IS A VIBRATION SPECTRUM (ALSO CALLED AN "FFT" OR "SIGNATURE")?

Most vibrations in the real world are complex combinations of various waveforms. *Figure 12 shows how the total waveform is actually made up of a series of smaller waveforms*, each of which correspond to an individual frequency (1X RPM, 2X RPM, 3X RPM, etc.). *Each of these individual waveforms will algebraically add to one another to generate the total waveform* which can be displayed either on an oscilloscope or on an analyzer.

One of the most important points to understand about the total time waveform is that it shows the total vibration motion of the machine or structure to which the vibration transducer is attached. If one can begin to comprehend this point, examination of the time waveform can go far in helping him diagnose both the cause and severity of problem(s) occurring within a machine.

However, particularly when an analyst is just beginning within the field of vibration analysis (typically less than 3 years full-time experience), displaying and using the time waveform can be very difficult and labor intensive if one needs to determine frequencies. To simplify the process, a Fast Fourier Transform (FFT) is generated and displayed within most of today's vibration data collectors and spectrum analyzers. *An FFT is a computer (microprocessor) transformation from time domain data (amplitude versus time) into frequency domain data (amplitude versus frequency).*

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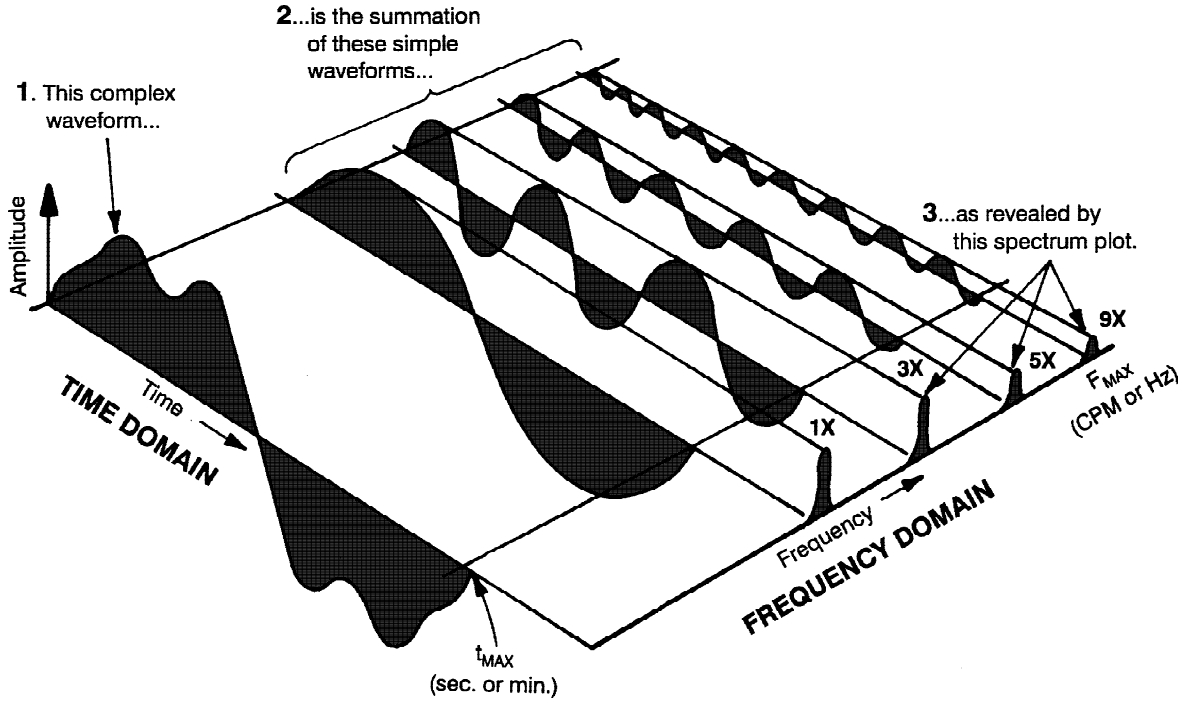

FIGURE 12
COMPARISON OF TIME & FREQUENCY DOMAINS

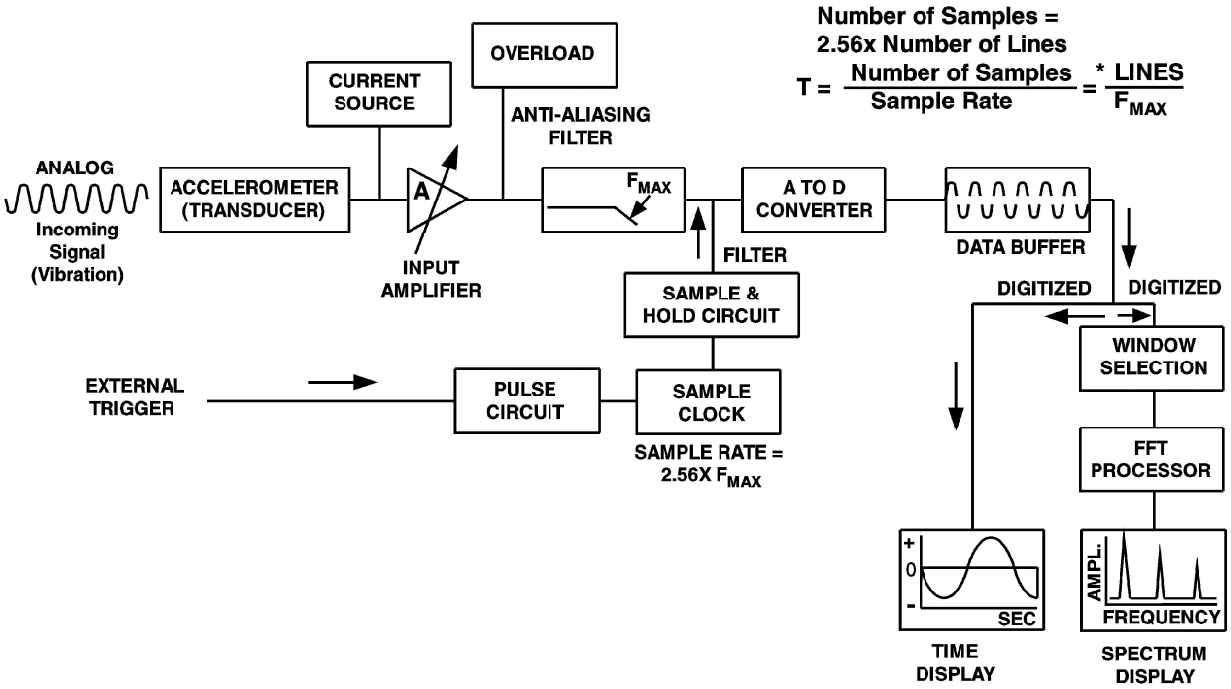

FIGURE 13
BLOCK DIAGRAM OF A GENERAL FFT ANALYZER TO SHOW HOW A DISPLAY IS PRODUCED IN EITHER THE TIME DOMAIN OR FREQUENCY DOMAIN

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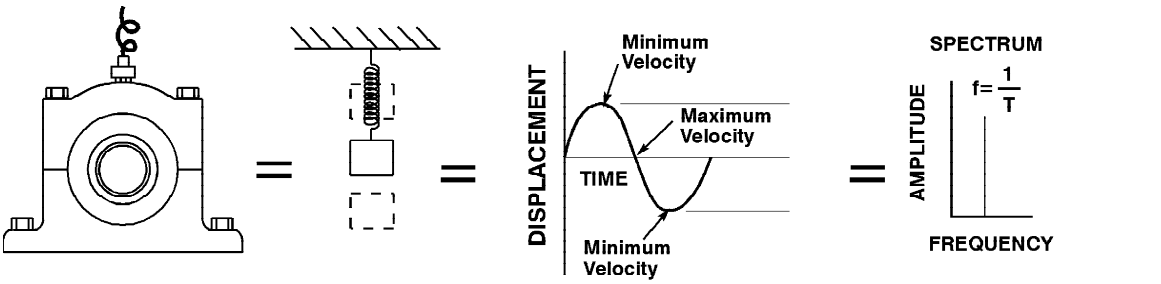
Figure 13 is a block diagram of a general FFT analyzer. Its purpose is to show how a digital time waveform or FFT spectrum is generated from the incoming analog vibration raw signal. The process on how an FFT spectrum is produced will be covered in later chapters. Remember that *an FFT is a microprocessor algorithm* (mathematical operation also used in computers) *applied to the incoming sampled data from the signal captured in the analog world* (time domain). This *must be transformed into the frequency domain using a series of mathematical operations.*

This FFT calculation technique was developed by Baron Jean Baptiste Fourier over 170 years ago (1822). Fourier proved that any real-world complex waveform can be separated into simple sinusoidal waveform components. The converse is also true: any series of simple sine waves can be combined to create the complex total waveform. As the sine waves are separated from the combined waveform, they are converted to vertical peaks which have an amplitude (as determined by their heights) and are given a position along the frequency axis. *This frequency domain presentation of a time waveform is called a spectrum (spectra, plural). A spectrum is also sometimes referred to as a "signature" or as an "FFT"* if an FFT analyzer is used.

Figure 14 summarizes the steps involved in capturing the total vibration waveform and transforming it into the frequency domain (FFT) as the signal is sent from a transducer mounted on the bearing housing in the real world. *Of course, this figure shows the transformation of only* one frequency. *In the real world, machines will generate many frequencies from many sources within the machine.* Diagnosing these spectra will be the main topics of the chapters in which the items in the *information will be summarized in an* "**Illustrated Vibration Diagnostics Chart**" developed by Technical Associates will be summarized.



(a) Real-world transducer mounted on vibrating part

(b) Spring mass system

(c) Time waveform generated by the transducer

(d) Frequency domain spectrum (FFT) of the time waveform (c)

FIGURE 14
STEPS IN THE CONVERSION OF A VIBRATION INTO AN FFT SPECTRUM

2.16 DIFFERENCE BETWEEN RMS, PEAK AND PEAK-TO-PEAK AMPLITUDE

Table I is a list of formulas which can be used to convert from one amplitude parameter to another. That is, it allows one to convert from displacement to velocity at a certain frequency; or from velocity to acceleration, etc. Thus, if one parameter is a peak value which has been measured, then the parameter which is being calculated will also be a peak value.

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TABLE I
CONVERSION FORMULAS FOR VARIOUS AMPLITUDE UNITS (Ref. 5)

ENGLISH	METRIC
D= 19,100 (V)/F	D_M= 19,100 (V_M)/F
D= 70,471,000 (A)/F^2	D_M= 1.79 X 10^9 (A)/F^2
V= DF/19,100	V_M= D_MF/19,100
V= 3690 (A)/F	V_M= 93,580 (A)/F
A= DF^2/70,471,000	A= D_MF^2/1.79 X 10^9
A= VF/3690	A= V_MF/93,580
A= Acceleration (g PK)	A= Acceleration (g PK)
V= Velocity (in/sec PK)	V= Velocity (mm/sec PK)
D= Displacement (mils PK-PK)	D= Displacement (microns PK-PK)
F= Frequency (CPM)	F= Frequency (CPM)

However, conversions from one vibration parameter to another are normally done by the software and electronics within the vibration instrument. Also, the electronics can perform all the necessary conversions for peak-to-peak, peak, and RMS (root-mean-square) amplitude values. Normally, Europeans use RMS velocity amplitudes, while Americans have adopted peak values even though, in reality, the instruments truly display RMS spectra, *and then electronically convert* them to so-called "peak" or "peak-to-peak" spectra by multiplying each of the amplitudes of each of the frequencies by 1.414 ($\sqrt{2}$) in the case of RMS-to-Peak; or by 2.828X in the case of RMS to "peak-to-peak" (assuming such waveform is sinusoidal). Figure 15 compares the English vibration units with the Metric.

VIBRATION CHARACTERISTICS	COMMON UNITS OF MEASUREMENT	
	ENGLISH	METRIC
FREQUENCY	CPM OR Hz	CPM OR Hz
DISPLACEMENT	MILS PEAK-TO-PEAK	MIICRONS, PEAK
VELOCITY	IN/SEC RMS IN/SEC PEAK	MM/SEC RMS MM/SEC PEAK
ACCELERATION	G PEAK	G RMS
PHASE	DEGREES	DEGREES

FIGURE 15
COMPARISON OF ENGLISH AND METRIC VIBRATION UNITS (Ref. 5)

Figure 16 shows how one unit of amplitude can be converted to another; that is, from RMS to peak, peak-to-peak, and vice versa. *These conversions apply only to pure sinusoidal waves only* (likely caused by almost pure unbalance), similar to the waveform shown in Figure 10.

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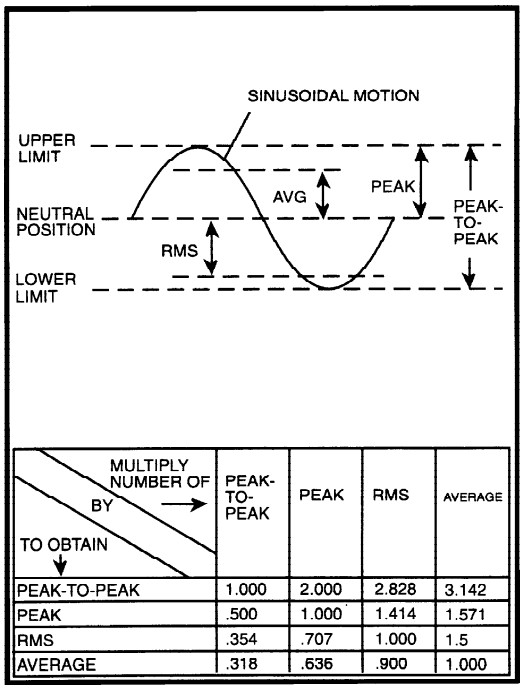

MULTIPLY NUMBER OF → / BY / TO OBTAIN ↓	PEAK-TO-PEAK	PEAK	RMS	AVERAGE
PEAK-TO-PEAK	1.000	2.000	2.828	3.142
PEAK	.500	1.000	1.414	1.571
RMS	.354	.707	1.000	1.5
AVERAGE	.318	.636	.900	1.000

FIGURE 16
COMPARISON OF PEAK, PEAK-TO-PEAK, RMS, AND AVERAGE FOR A PURE SINUSOIDAL TIME WAVEFORM

2.17 WHEN TO USE DISPLACEMENT, VELOCITY, OR ACCELERATION

Displacement is normally thought to be the most useful vibration parameter in frequency ranges less than approximately 600 CPM (10 Hz). However, a frequency must be used along with displacement to evaluate vibration severity as shown by Figure 17. For instance, 2 mils Pk-Pk of vibration at 3600 CPM is much more destructive than is the same 2 mils vibration at 300 CPM (see Figure 17 which is a displacement and velocity severity chart developed years ago for "general rotating machines"). Thus, *displacement alone is unable to evaluate vibration severity throughout the entire frequency range (even for a low-speed machine)*.

Acceleration is also frequency dependent (see Figure 18 which is a vibration severity chart for acceleration). Typically, *acceleration is recommended for use when sources within a machine generate frequencies over approximately 120,000 CPM (2000 Hz)*. These sources may include gear mesh frequencies (#teeth X RPM) and blade passing frequencies (#blades X RPM) for high-speed centrifugal as well as harmonics (or multiples) of these frequencies.

On the other hand, *velocity is not nearly so frequency dependent in the frequency range from approximately 600 - 120,000 CPM (10 - 2000 Hz)*. *Even when vibration frequencies are generated from 300 to as high as 300,000 CPM in a machine, velocity is usually the unit of choice* (although one will have to take into account the roll-off in sensitivity of velocity at frequencies exceeding 120,000 CPM as shown in Figure 19). For example, if one allowed a velocity of .10 in/sec at 120,000 CPM for a fault such as a gear mesh frequency, he would likely allow only a level of about .04 in/sec at a frequency of 300,000 CPM [(120,000/300,000)(.10) = .04] as per the equations and graphs identified as "CONTOURS OF EQUAL SEVERITY" shown in Figure 19. For the same reason, if one allowed a velocity of .314 in/sec. at 600 CPM, he should allow a level of only .031 in/sec. at a frequency 60 CPM due to fall off in velocity below 600 CPM (10Hz) as shown by Figure 19. Figure 19 shows that if he still allowed .314 in/sec. at 60 CPM, this would be

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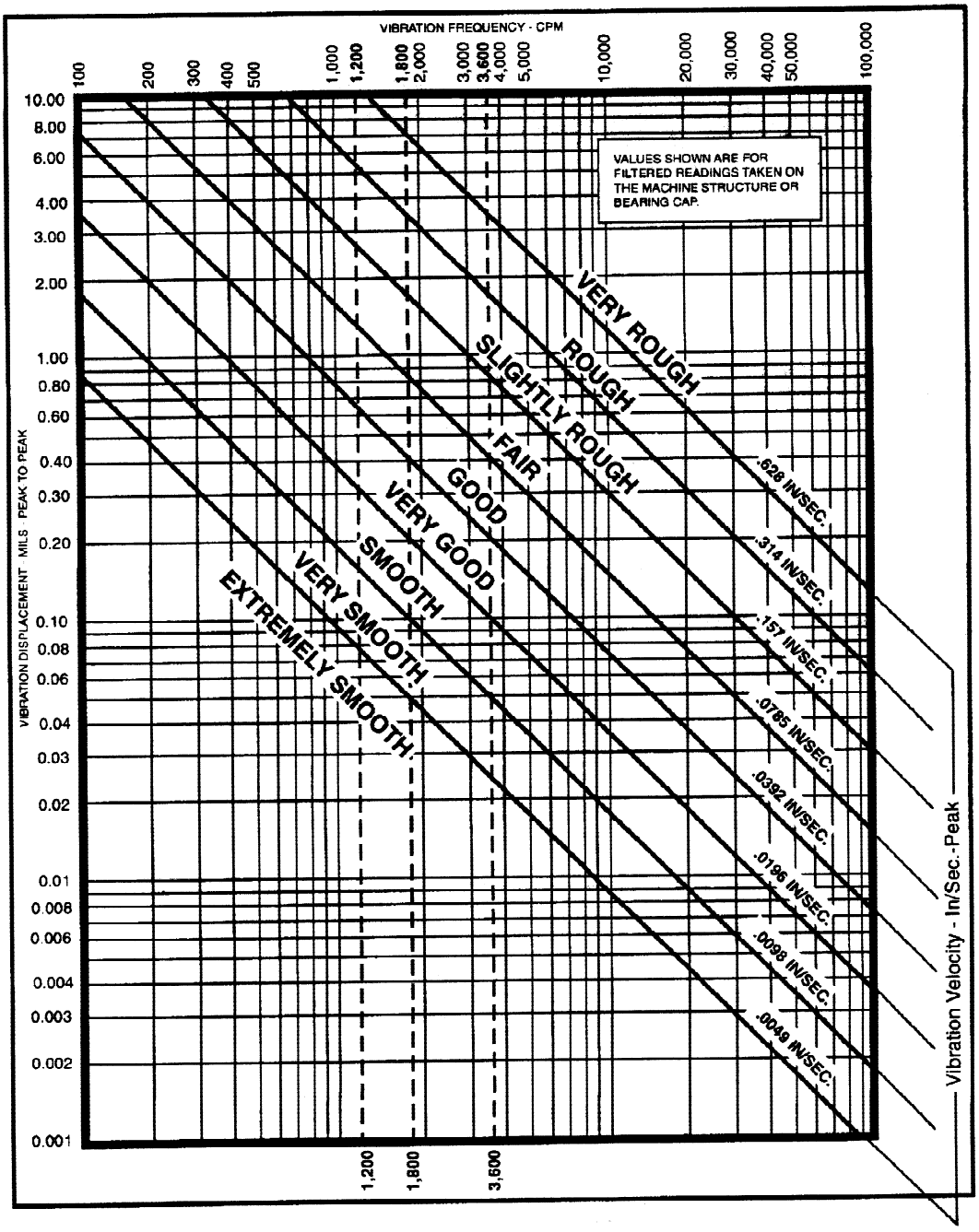

FIGURE 17
VIBRATION DISPLACEMENT & VELOCITY SEVERITY CHART FOR GENERAL HORIZONTAL ROTATING MACHINERY
(Source: Entek IRD International, Milford, Ohio)

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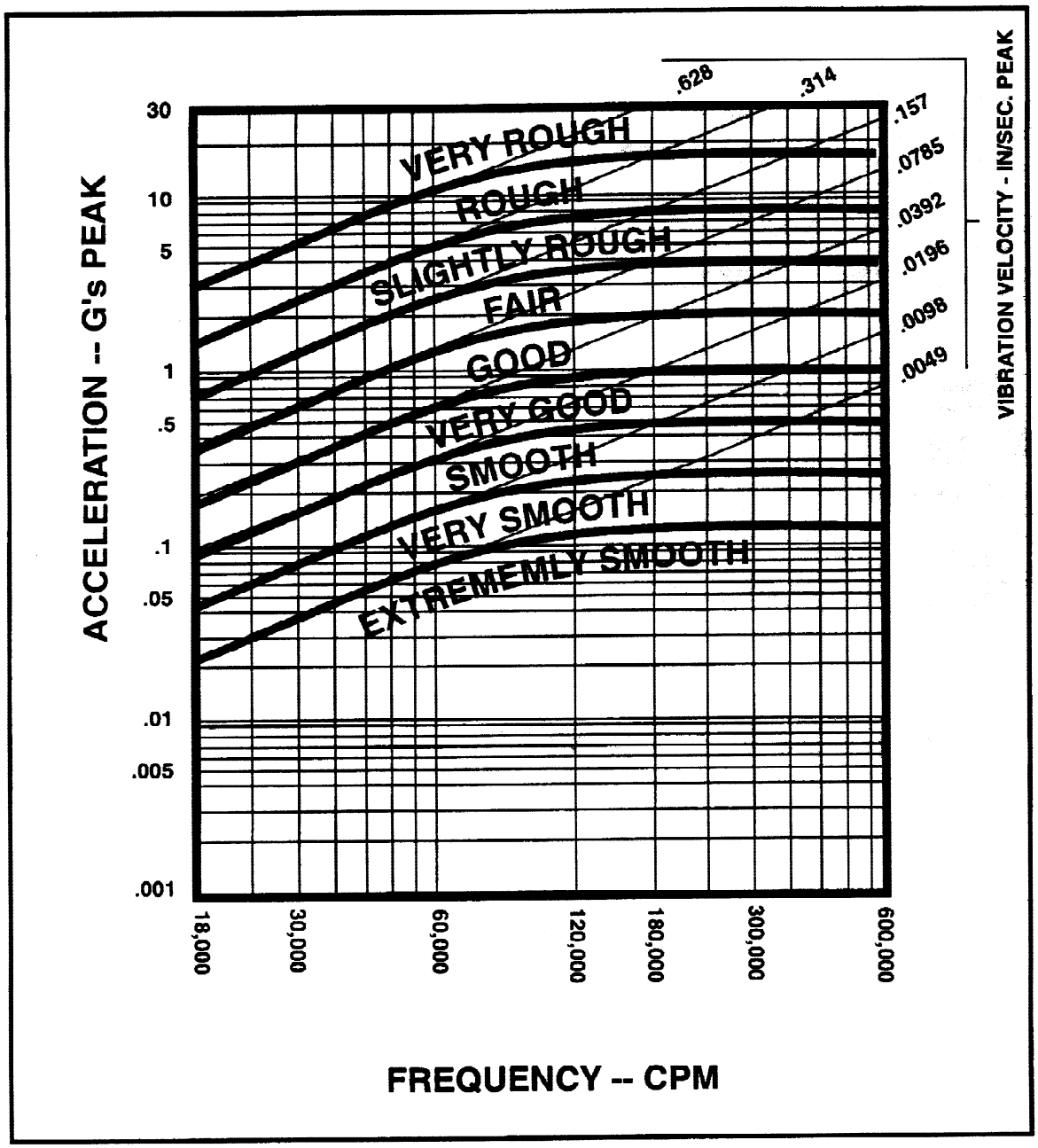

FIGURE 18
VIBRATION ACCELERATION & VELOCITY SEVERITY CHART FOR GENERAL HORIZONTAL ROTATING MACHINERY
(Source: Entek IRD International, Milford, Ohio)

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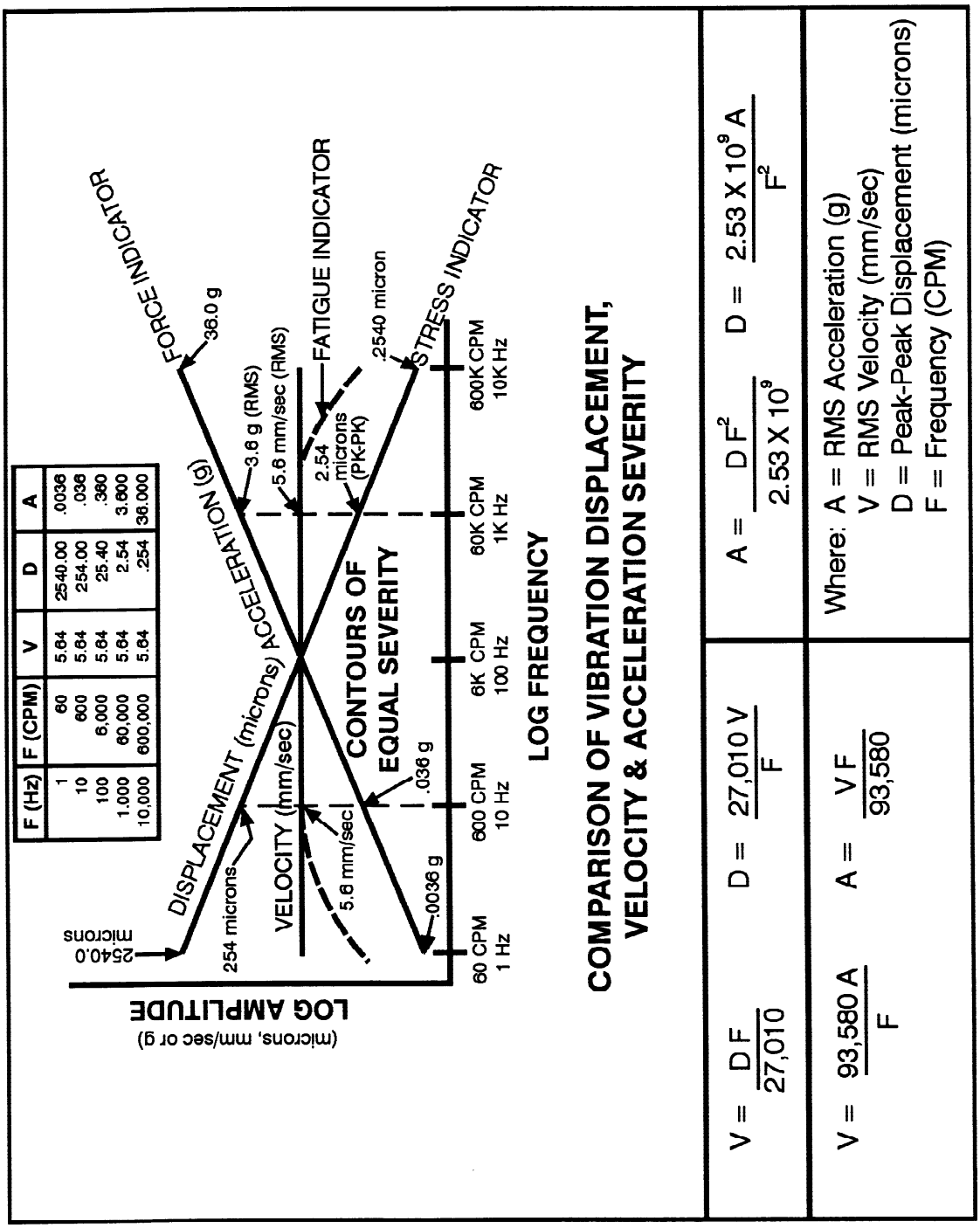

FIGURE 19

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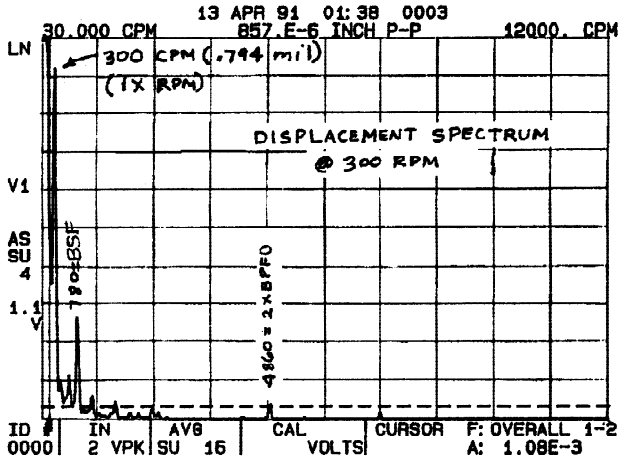
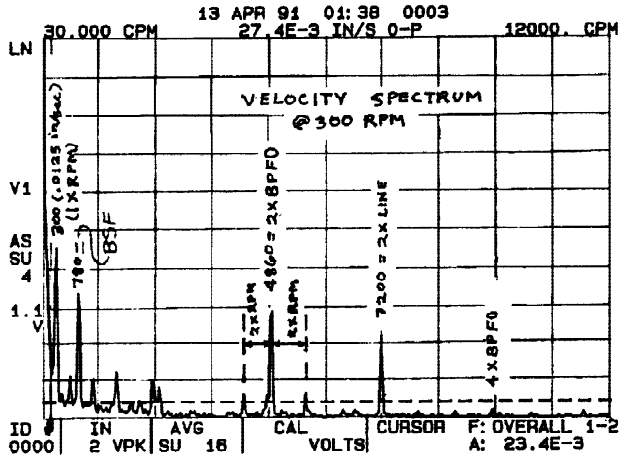
A.



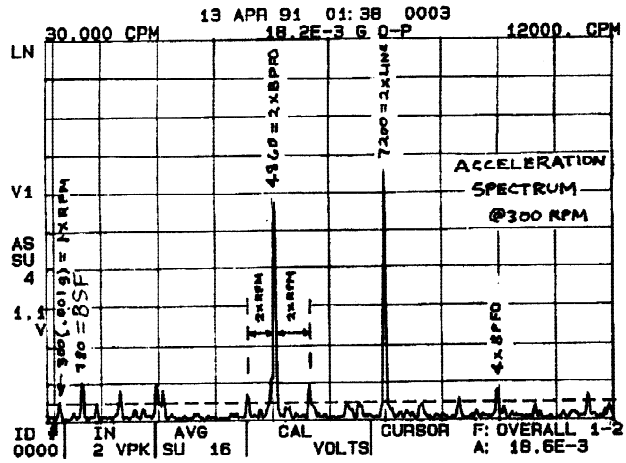
INCH P-P	CPM
794.E-6	300.00
82.2E-6	420.00
97.8E-6	600.00
227.E-6	780.00
50.9E-6	1110.0
39.1E-6	1589.9
31.3E-6	4860.0

B.



IN/S 0-P	CPM
12.5E-3	300.00
1.75E-3	420.00
3.13E-3	600.00
9.26E-3	780.00
1.13E-3	900.00
2.88E-3	1110.0
1.13E-3	1500.0
3.38E-3	1589.9
1.13E-3	1890.0
1.25E-3	2070.0

C.



G 0-P	CPM
997.E-6	300.00
1.91E-3	780.00
1.50E-3	1589.9
1.75E-3	2340.0
1.50E-3	2490.0
1.33E-3	4260.0
10.5E-3	4860.0
1.83E-3	5579.7
914.E-6	6360.0
914.E-6	6629.7
11.9E-3	7200.0

FIGURE 20
COMPARISON OF DISPLACEMENT, VELOCITY & ACCELERATION SPECTRA ON A 300 RPM FAN WITH BEARING PROBLEMS

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equivalent to an excessive displacement of 100 mils (PK-PK). Velocity can still be used at low speeds like 60 CPM, since bearing defect frequencies, gear mesh frequencies and their harmonics will still normally be higher than 600 CPM (10Hz), which is considered the "break point" between low speed analysis and moderate speed analysis techniques.

2.171 What is the Advantage of Using Velocity?

Figure 19 shows the consistency which velocity has over a wide, flat frequency range as compared with displacement and acceleration. They tend to favor the low and the high ends of the frequency scale, respectively. Note in Figure 19 that all 3 amplitude parameters are displayed on the same graph, using the .314 in/sec peak velocity amplitude as a basis for the calculation of the "CONTOURS OF EQUAL SEVERITY".

The following example, as displayed in Figure 20, shows 3 spectra in (A) displacement, (B) velocity, and (C) acceleration of the same waveform. Carefully analyze these spectra for a possible bearing defect problem. Although the 1X RPM peak (300 CPM) appears in all three spectra and is even quite outstanding in Figure 20A, it is not the most significant problem here.

What happens to the display of bearing defect frequencies if using one vibration parameter or another is possibly more important when one is evaluating machine condition. Whether or not the analyst will see these important bearing frequencies in his spectra may depend upon his choice of amplitude parameter. While bearing frequencies at 4860 CPM and 9720 CPM with their sideband frequencies are clearly seen in Figures 20B and 20C, note that the frequency at 9720 CPM is *missed entirely* in the displacement spectrum, as well as the sidebands surrounding the 4860 CPM frequency in Figure 20A. This is very important. If an acceleration spectrum in Figure 20C wasn't taken, these frequencies still showed up significantly in the Figure 20B velocity spectrum. If the displacement spectrum in Figure 20A wasn't taken, the 1X RPM spike was still significant in the velocity spectrum of Figure 20B. Therefore, if only a velocity spectrum had been taken, as in Figure 20B, both types of problems would be clearly visible.

Thus, it is important to note that *a velocity spectrum has a much wider usable frequency range than do spectra in displacement or acceleration. Combining this characteristic with velocity's direct relationship to vibration severity makes velocity the best measurement parameter to use for most rotating machinery*. This is especially true when frequencies are below 120,000 CPM (2000 Hz).

2.18 HOW MUCH IS TOO MUCH VIBRATION?

Through the years, the general vibration severity chart of Figure 17 has been commonly used. *However, this chart was never intended to be used on all machine types and configurations to choose vibration limits to give adequate warning of existing or impending problems. To help meet this need, Technical Associates has developed a comprehensive vibration severity chart, shown in Figure 21, which is entitled "CRITERIA FOR OVERALL CONDITION RATING".* This chart applies to a wide variety of machines over a wide range of operating speeds from 600 - 60,000 RPM. *These levels are peak overall velocity levels (in/sec).* They were acquired through many years of actual vibration data acquisition on a diverse array of machine types. The columns entitled "GOOD" and "FAIR" are used to give a machine an "OVERALL CONDITION RATING" *based on the highest overall vibration level found on any one of the machine measurement points*. In general, machines allowed to operate above "ALARM 1" will likely fail prematurely if problems are not identified and corrected. "ALARM 2" levels are 50% higher than those of "ALARM 1". *If machines are allowed to operate above "ALARM 2", they may suffer catastrophic failure if left unaddressed.*

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This Technical Associates Rating Chart does not cover all types of machines. *Further, it is meant for "IN-SERVICE" equipment only. It is not meant to be used for "ACCEPTANCE TESTING"*. For machines not included in the chart of Figure 21, one could use the Figure 17 severity chart *or a statistical method to develop other alarm levels. A statistical comparison can be conducted if the machines are similar* in construction, drive configuration, operating speeds, loading and in internal components. *This statistical method is especially effective when several surveys on the machines have been conducted. It is practical to revise the alarms since the original vibration levels are almost always reduced as machine problems and defects are corrected.*

Shock Pulse, HFD, and Spike Energy, as other measurement parameters, will be discussed later in several other chapters.

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FIGURE 21

Metric Version Of Table II

TABLE II

TECHNICAL ASSOCIATES OF CHARLOTTE, P.C.

CRITERIA FOR OVERALL CONDITION RATING (RMS OVERALL VELOCITY, mm/sec.)*

1. Assuming Machine Speed = 600 to 60,000 RPM.
2. Assuming Measurements by Accelerometer or Velocity Pickup securely mounted as Close as Possible to Bearing Housing.
3. Assuming Machine Is Not Mounted on Vibration Isolators (for Isolated Machinery, Set Alarm 30% - 50% Higher).
4. Set Motor Alarms the Same as that for the Particular Machine Type unless Otherwise Noted.
5. Set Alarms on Individual External Gearbox Positions about 25% Higher than that for a particular Machine Type.

MACHINE TYPE	GOOD	FAIR	(WARNING) ALARM 1	(FAULT) ALARM 2
COOLING TOWER DRIVES				
Long, Hollow Drive Shaft	0 - 7.0	7.0 - 11.0	11.0	16.0
Close Coupled Belt Drive	0 - 5.0	5.0 - 7.5	7.5	12.0
Close Coupled Direct Drive	0 - 3.5	3.5 - 5.5	5.5	8.0
COMPRESSORS				
Reciprocating	0 - 6.0	6.0 - 9.0	9.0	13.5
Rotary Screw	0 - 5.5	5.5 - 8.0	8.0	12.0
Centrifugal With or W/O External Gearbox	0 - 3.5	3.5 - 5.5	5.5	8.0
Centrifugal - Integral Gear (Axial Meas.)	0 - 3.5	3.5 - 5.5	5.5	8.0
Centrifugal - Integral Gear (Radial Meas.)	0 - 2.5	2.5 - 4.5	4.5	6.5
BLOWERS (FANS)				
Lobe-Type Rotary	0 - 5.5	5.5 - 8.0	8.0	12.0
Belt-Driven Blowers	0 - 5.0	5.0 - 7.5	7.5	11.5
General Direct Drive Fans (with Coupling)	0 - 4.5	4.5 - 6.5	6.5	10.0
Primary Air Fans	0 - 4.5	4.5 - 6.5	6.5	10.0
Large Forced Draft Fans	0 - 3.5	3.5 - 5.5	5.5	8.0
Large induced Draft Fans	0 - 3.0	3.0 - 5.0	5.0	7.0
Shaft-Mounted Integral Fan (Extended Motor Shaft)	0 - 3.0	3.0 - 5.0	5.0	7.0
Vane-Axial Fans	0 - 2.5	2.5 - 4.5	4.5	6.5
MOTOR/GENERATOR SETS				
Belt-Driven	0 - 5.0	5.0 - 7.5	7.5	12.0
Direct Coupled	0 - 3.5	3.5 - 5.5	5.5	8.0
CHILLERS				
Reciprocating	0 - 4.5	4.5 - 7.0	7.0	10.5
Centrifugal (Open-Air) - Motor & Comp. Separate	0 - 3.5	3.5 - 5.5	5.5	8.0
Centrifugal (Hermetic) - Motor & Impellers Inside	0 - 2.5	2.5 - 4.0	4.0	6.0
LARGE TURBINE/GENERATORS				
3600 RPM Turbine/Generators	0 - 3.0	3.0 - 5.0	5.0	7.0
1800 RPM Turbine/Generators	0 - 2.5	2.5 - 4.0	4.0	6.0
CENTRIFUGAL PUMPS				
Vertical Pumps (4m - 6m Height) } Height from Grade to Top Motor Bearing.	0 - 6.0	6.0 - 9.0	9.0	13.5
Vertical Pumps (3m - 4m Height) } May be Necessary to Spec. Lower Alarm	0 - 5.0	5.0 - 7.5	7.5	11.5
Vertical Pumps (2m - 3m Height) } for Lower Motor Bearing & for Upper	0 - 4.0	4.0 - 6.0	6.0	9.5
Vertical Pumps (0m -2m Height) } Pump Bearing (depending on height).	0 - 3.5	3.5 - 5.5	5.5	8.0
General Purpose Horizontal Pump - Direct Coupled	0 - 3.5	3.5 - 5.5	5.5	8.0
Boiler Feed Pumps - Horizontal Orientation	0 - 3.5	3.5 - 5.5	5.5	8.0
Hydraulic Pumps - Horizontal Orientation	0 - 2.0	2.0 - 3.5	3.5	5.5
MACHINE TOOLS				
Motor	0 - 1.2	1.2 - 1.8	1.8	2.7
Gearbox Input	0 - 1.8	1.8 - 2.7	2.7	4.0
Gearbox Output	0 - 1.2	1.2 - 1.8	1.8	2.7
Spindles:				
a. Roughing Operations	0 - .70	.70 - 1.2	1.2	1.8
b. Machine Finishing	0 - .45	.45 - .72	.72	1.1
c. Critical Finishing	0 - .27	.27 - .45	.45	.72

***NOTE:The "ALARM 1" and "ALARM 2" overall levels given above apply only to in-service machinery which has been operating for some time after initial installation and/or overhaul. They do not apply (and are not meant to serve as) Acceptance Criteria for either new or rebuilt machinery.**

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2.2 OVERVIEW OF THE STRENGTHS AND WEAKNESSES OF TYPICAL VIBRATION INSTRUMENTS

2.21 INTRODUCTION

The purpose of vibration instrumentation is to accurately measure vibration amplitudes, frequencies, and phase so that a reliable determination of a machine's condition can be made. There are 5 basic types of vibration instruments as follows:

1) **Overall Level Vibration Meters**

2) **Swept-Filter Analyzers**

3) **FFT Programmable Data Collectors**

4) **Real-Time Spectrum Analyzers**

5) **Instrument Quality Tape Recorders**

2.22 INSTRUMENT COMPARISONS

This section is meant to evaluate the general capabilities of the 5 types of instruments listed above. It is important to note that not all makes and models of these instrument types which are configured to exhibit each of these characteristics that will be featured. However, this section provides a good checklist to review with an instrument manufacturer to fully understand the instrument's capabilities.

Table II will present each type of instrument and list the general capabilities each possesses. The comparison characteristics will be defined in more detail below only if they are somewhat complex.

A) **Portability -** Can the equipment be easily carried around the plant or mill? How much does it weigh?

B) **Typical Frequency Range -** Describes the typical range of frequencies from a low limit to a high limit in which an instrument of each particular type can accurately measure according to a specified amplitude tolerance (usually ± 10% or ± 3dB).

C) **Data Measurement Format -**
   1) OL (Overall Level)
   2) SF (Swept Filter)
   3) FS (Frequency Spectrum)
   4) TWF (Time Waveform)

D) **Typical Display Types -**
   1) LCD (Liquid Crystal Display)
   2) MS (Monochrome Screen)
   3) AM (Analog Meter)

E) **Typical Transducer Types -**
   1) A (Accelerometer)
   2) V (Velocity Transducer)
   3) P (Proximity Eddy Current Probe)

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F) **Phototach and/or Strobe Light Capabilities -** Can the instrument type normally use a Phototach or strobe light with which it can measure phase, as well as enabling it to possibly perform operating deflection shape, modal analysis and/or synchronous time averaging?

G) **Multi-Channel Availability -** Is this instrument type typically available in more than one channel?

H) **Spike Energy, HFD, or SPM (Shock Pulse Measurement) Capability -** Can the instrument type typically measure one of these parameters?

I) **High Frequency Enveloped Spectral Measurement Capability -** High frequency enveloped spectra are known by different vendors as "Spike Energy Spectra[11]", "Amplitude Demodulated Spectra[12]", or "Acceleration Enveloped Spectra[13]", which are usually measured in the 5000 - 50,000 Hz (300,000 - 3,000,000 CPM) frequency range. However, the "SEE" spectrum[14] developed by SKF Condition Monitoring is measured in the 250,000 - 350,000 Hz (15,000,000 - 21,000,000 CPM) range (where SEE refers to Spectral Emitted Envelope). These parameters will be covered in later chapters of the text.

J) **Spectral Display Update -** How fast does the screen refresh itself with up-to-date data?
- **LT (Live Time) -** Screen updates every 1 to 4 seconds depending on the instrument model and the settings, such as the frequency span, the number of lines of resolution, the overlap processing percentage, etc.
- **RT (Real Time) -** Screen updates almost instantaneously, particularly in higher frequency spans (again depends on instrument setup parameters just as on a data collector).

K) **Ease of Use -** An assessment rating from simple to complex based on the time and training normally required to operate the instrument effectively. The assessment has to include a consideration of whether the instrument will be used regularly (daily, weekly) or occasionally (monthly).

L) **Time Waveform Storage Capability -** Can this instrument type typically acquire and store time waveform?

M) **Frequency Spectra Storage Capability -** Can this instrument type typically acquire and store frequency spectra?

N) **Predictive Maintenance (PMP) Software Compatibility -** Is the instrument compatible with available condition monitoring software to set up overall and spectral alarms, trend data, routes, etc.?

O) **Natural Frequency Testing Capability -** Can the instrument be used to conduct "bump" or "impulse" tests, coastdown/runup tests, Bode' or Polar plot measurements?

P) **ODS (Operating Deflection Shape) Capability -** The ability to simultaneously measure the amplitude and phase at a particular forcing frequency (such as 1X or 2X RPM) are measured at specified locations on a structure or machine and typically are downloaded into a personal computer. Software in the computer is designed to produce animated operating deflection shape plots on the screen. This will simulate how the remainder of the machine or structure is moving in relation to one of the points. This can be accomplished using a single channel analyzer, along with a once/revolution trigger.

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Q) **Experimental Modal Analysis Capability -** The capability to measure items required by modal analysis such as natural frequencies, mode shapes, coherence and transfer functions. Modal analysis involves exciting the natural frequencies of the structure with the use of a "modal hammer" (force transducer) or by a shaker likewise outfitted with a force transducer, and measuring the response with an accelerometer. This analysis requires at least a two channel analyzer.

R) **STA (Synchronous Time Averaging) Capability -** The capability to eliminate all frequencies that are not exact multiples of a designated frequency. The spectrum being measured will be limited to only multiples of the fundamental frequency (most often, operating speed) which is synchronous with the trigger source (such as a phototach or strobe light). The nonsynchronous frequencies will disappear from the spectrum and time waveform if a sufficient number of averages are taken (often 250 to 500 averages).

S) **Waterfall or Cascade Plotting Capability -** The capability to display one FFT after another during a "runup" or "coastdown" and/or from one PMP survey to the next on the screen.

T) **Relative Costs -** From a basic instrument "low-end" cost, the range may vary considerably to the "high-end" cost, depending on the software, cabling, number of channels, the auxiliary equipment, and other "extras" to be purchased. The "nominal cost" represents what is normally paid for these instruments and the necessary "extras".

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TABLE II
TYPICAL VIBRATION MEASUREMENT INSTRUMENT CHARACTERISTICS @ 6/93

A. PORTABILITY Approximate Weight (kg)	YES 0.5 - 1	SEMI 15-25	YES 2.5 - 5	SEMI 10 - 30	SEMI 10 - 20
B. TYPICAL FREQUENCY RANGE (CPM) Low End High End	 120 600,000	 60 600,000	 60 2,400,000	 0 6,000,000	 VARIES VARIES
C. DATA MEASUREMENT FORMAT	OL	OL, SF	OL, FS, TWF	OL, FS, TWF	-
D. TYPICAL DISPLAY TYPES	AM	AM	LCD	MS TYPICALLY	-
E. TYPICAL TRANSDUCER TYPES	PER MANUF	A,V	A, V, P	A, V, P	A, V, P
F. PHOTOTACH AND/OR STROBE LIGHT	NO	YES	MOST USE BOTH	P'TACH-YES STROBE-NO	P'TACH-YES STROBE-NO
G. MULTI-CHANNEL AVAILABILITY	NO	LIMITED	SOME	YES	YES
H. SPIKE ENERGY, HFD, OR SHOCK PULSE	SOME	SOME	MOST	NO	NO
I. HIGH FREQUENCY ENVELOPED SPECTRUM	NO	NO	MOST	FEW	NO
J. SPECTRAL DISPLAY UPDATE	-	-	LT	RT	-
K. EASE OF USE	SIMPLE	SIMPLE	FAIRLY SIMPLE	DEMANDING	VARIES
L. TIME WAVEFORM STORAGE CAPABILITY	NO	HC ONLY	MOST	YES	YES
M. FREQUENCY SPECTRA STORAGE	NO	NO	YES	YES	-
N. PREDICTIVE MAINT. SOFTWARE AVAILABILITY	NO	NO	YES	SOME	-
O. NATURAL FREQUENCY TESTING CAPABILITY	NO	LIMITED	GOOD	SUPERIOR	-
P. OPERATING DEFLECTION SHAPE (ODS)	NO	NO	YES	SOME	-
Q. EXPERIMENTAL MODAL ANALYSIS CAPABILITY	NO	NO	FEW	MOST	-
R. SYNCHRONOUS TIME AVERAGING (STA)	NO	NO	SOME	YES	-
S. WATERFALL PLOTTING CAPABILITY	NO	NO	FEW	MOST	-
T. RELATIVE COSTS (US $) Low End High End Nominal Expected Cost	 $400 $2,800 $1,200	 $1,000 $9,000 $6,000	 $3,500 $18,000 $9,000	 $4,000 $40,000 $15,000	 $5,000 $100,000 $15,000

ABBREVIATIONS

A - ACCELEROMETER
AM - ANALOG METER
FS - FREQUENCY SPECTRA
HC ONLY - PRINTED HARD COPY ONLY
LCD - LIQUID CRYSTAL DISPLAY
LT - LIVE TIME (TYPICALLY UPDATED EVERY 1-4 SECONDS)
V - VELOCITY TRANSDUCER
MS - MONOCHROME DISPLAY
OL - OVERALL LEVEL
P - PROXIMITY PROBE
RT - REAL-TIME (HIGH SPEED SCREEN REFRESH RATES)
SF - SWEPT FILTER
TWF - TIME WAVEFORM

NOTE: THE ABOVE TABLE HAS NOT TAKEN INTO ACCOUNT ALL OF THE VARIOUS INSTRUMENTS AVAILABLE ON THE MARKET TODAY. DUE TO THE RAPID ADVANCES IN TECHNOLOGY, MANY PIECES OF EQUIPMENT CANNOT BE CLEARLY DEFINED INTO THE ABOVE CATEGORIES. THIS TABLE IS ONLY MEANT TO BE USED AS AN EXAMPLE OF THE CAPABILITIES A USER WOULD EXPECT TO FIND FROM A PARTICULAR INSTRUMENT TYPE. THE ABOVE CHARACTERISTICS SHOULD BE DISCUSSED IN DETAIL WITH A PARTICULAR MANUFACTURER BEFORE PURCHASING.

NOTE: "CHARACTERISTIC" IS DEFINED FOR EACH LISTED ITEM IN SECTION 3.1. DISCUSSIONS COVERING THE GENERAL CAPABILITIES AND LIMITATIONS O EACH INSTRUMENT TYPE FOLLOW IN SECTION 3.2.

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2.23 GENERAL CAPABILITIES OF EACH VIBRATION INSTRUMENT TYPE

The following is a summary of the instruments covered previously in Table II outlining their major advantages and drawbacks.

2.231 Overall Level Vibration Meters

As the name implies, these instruments measure overall vibration (and some meters likewise measure Spike Energy, or another of the so-called ultrasonic bandpass parameters). Overall vibration refers to the overall or total amplitude summation of all the vibration in the form of acceleration, velocity, displacement, or one of the high frequency bandpass filtered parameters. At one time, these lightweight, portable instruments were used extensively, but (because of their limitations) have been replaced today by FFT Programmable Data Collectors. Some of the major drawbacks in using these instruments is their inability to display or store either spectra or time waveforms; their limited frequency ranges in most cases; and the requirement by most such meters that the vibration reading must be manually recorded which is cumbersome and time-consuming.

2.2311 Drawbacks in Measuring Only Total or Overall Vibration

For precision machinery, or for machinery that is critical to a plant's operation, routine overall vibration level recording is not sufficient. Numerous failures can occur with only a minute increase (or decrease) in the overall level, particularly if only displacement, velocity, or acceleration is used. This can occur if the problem is bearing wear, gear wear, cracked gear teeth, cracked rotor bars, etc. Even if a user feels that he can evaluate machine health by monitoring bearing condition using overall spike energy, HFD, or shock pulse, he should be aware that rolling element bearing condition is not the only cause for a high reading. Lubrication, cavitation, high pressure steam or air, gear condition, rotor rub, and belt squeal also can cause an increase in these high frequency bandpass parameters. To determine *what* is causing this parameter to increase will require a spectrum analysis or a high frequency enveloped spectrum measurement.

2.232 Swept-Filter Analyzers

These analyzers use a constant percentage analog filter (typically 2% - 10%) to sweep through a frequency range (typically from about 60 - 600,000 CPM, or 1 - 10,000 Hz). Thus, if using a 10% filter and measuring at 1000 CPM, the filter would include vibration from 950 to 1050 CPM. On the other hand, when up at a frequency of 100,000 CPM, the filter would include 95,000 to 105,000 CPM vibration. These analyzers have been replaced for the most part by FFT Programmable Data Collectors because these data collectors are capable of storing data and producing better resolved frequency spectra. However, swept-filter analyzers can be still used for field balancing, strobe light "slow motion studies" and phase analysis. A drawback is that the operator has to be near the instrument to use the strobe and tune in to the frequency. Some other drawbacks are they are not easily transported since they are fairly large and heavy in size and are too cumbersome to be used on a predictive maintenance route. Besides, the data cannot be stored by these analyzers, nor are they capable of storing or displaying time waveforms.

The operator has to be aware that swept-filter analyzers only capture events which occur when the filter happens to be measuring at the event's (i.e., transient vibration spike) frequency setting at a particular instant in time. The frequency precision is limited by the filters used. In all cases, the frequency resolution is limited to the filters used.

2.233 FFT Programmable Data Collectors

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FFT Programmable Data Collectors are the current state-of-the-art instruments of choice for predictive maintenance programs. The FFT capability transforms the time waveforms captured by these units into frequency spectra and most data collectors can display them on a small LCD screen in "live-time". Although the data collector was designed to collect data on many pieces of machinery, many of them can also be used as an analyzer in the field due to their graphics, FFT, live-time capabilities, and their ability to display time waveforms. Most of them can measure phase with the attachment of a strobe light or phototach and can also measure high frequency parameters such as HFD and Spike Energy. Some can also measure high frequency enveloped spectra, such as "Spike Energy spectra[11]", "Amplitude Demodulated spectra[12]", "Acceleration Enveloped spectra[13]" or "SEE spectra[14]".

While most of the data collectors are single channel instruments, others have anywhere from 2 to 4 channels. Only a few FFT data collectors are capable of multi-channel data input which is needed for modal analysis and helpful in operating deflection shape tests.

Some data collectors have the ability to measure 3 different parameters (i.e., velocity, acceleration, and spike energy) simultaneously with one push of the "store" button. Also, if a triaxial accelerometer is used, multi-channel units having at least 3 channels can display the spectra from all 3 directions simultaneously, with little or no loss in analyzer processing speed.

The frequency range of the average data collector is normally from 60 to 1,500,000 CPM (1 to 25,000 Hz), but some data collectors are now available with frequency measuring capabilities as low as 6 CPM (0.1 Hz), and as high as 360,000 CPM (6000 Hz). *With the intense development of data collectors going on today, no telling what their capabilities might be only within the next 5 to 10 years.*

2.234 Real-Time Spectrum Analyzers

The real-time spectrum analyzer is the most powerful diagnostic tool for advanced diagnostic techniques on the market. The "real-time" display updates "quicker than the eye" when the frequency span and other setup parameters are properly specified, as opposed to the "live-time" display in data collectors. In addition many of them have a built-in time buffer which allows one to store "runups" or "coastdowns" and play them back over and over again (similar to using a tape recorder). They also can capture short duration transient (less than 20 milliseconds) events and examine the data looking for potential problems.

Real-time analyzers are excellent in performing impulse natural frequency tests, coastdown/runup tests, and transient capture due to their "peak hold" capabilities. They can also generate Bode' and Polar plots to verify the location of natural frequencies. The multi-channel capabilities available in many of these units provide an excellent facility to capture data in operating deflection shape and modal analysis. Most are capable of performing synchronous time averaging and order tracking. Phototach input is available for phase analysis (or multi-channel RTA's can use another accelerometer or force transducer as a reference for phase measurement).

Since a real-time spectrum analyzer is normally complex, the user will require additional training and frequent use to remain proficient. Also, it is not usually very portable. Some are now equipped with a 3.5 inch (1.44 Mb) floppy drive (or even large megabyte hard drives) which provide a virtually limitless storage capacity for spectral and time waveform data. Recent RTA's have built-in computers with special cards which allow PMP software to be installed on them. Some are also equipped with word processing, spreadsheets, and graphics software to provide "on-the-spot" report generation.

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The typical real-time analyzer today is capable of measuring frequencies in a very wide range from 0 - 6,000,000 CPM (0 - 100,000 Hz). In summary, the real-time analyzer can prove invaluable for experienced analysts who want to conduct sophisticated diagnostic investigations. However, with the constantly increasing capabilities of today's data collectors, they are no longer an absolute necessity for a "complete" condition monitoring program.

2.235 Instrument Quality Tape Recorders

Instrument quality tape recorders can simultaneously record many different signals (whether from vibration pickups, pressure transducers, tachometers, current transformers, phototachs, etc.). Tape recorders can capture short-lived transient events which cannot even be "seen" by an analyzer. After a signal has been captured on tape, it can be analyzed back in the office at a much lower speed to allow capture of short-lived transients, particularly since it may be played back over and over again.

There are two types of tape recorders available today: analog and digital. The analog type records the actual signal input from the transducer without breaking it into a number of sampled points. However, their dynamic range is limited to *40 - 48 dB*. On the other hand, *digital tape recorders typically have a dynamic range of about 72 - 80 dB*. Therefore, they are about 30 dB more amplitude sensitive. That is, *digital recorders can detect small amplitude frequencies with amplitudes over 30 times lower* than can analog devices in the simultaneous presence of much higher amplitude frequencies. Incidentally, the digital tape recorder samples the input signal at a specified rate and reproduces it as stored numbers.

Tape recorders accurately record the time waveform of the vibration which can be analyzed later with a real-time spectrum analyzer or data collector. Both types of tape recorders have multi-channel capabilities (up to 64 channels or more) which allow the capture of many data points simultaneously.

However, instrument quality tape recorders can be very costly. As an alternative, they can be rented for use. The price then will depend on the frequency range, the number of channels required, and the type. Furthermore, they may be somewhat complicated to use, so frequent use to remain proficient may be required.

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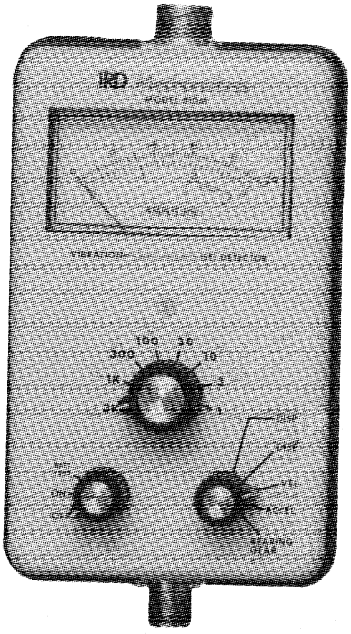
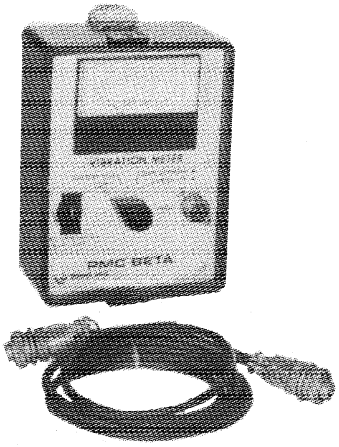
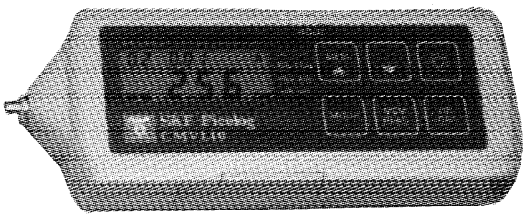


A. IRD 810 VIBRATION SPIKE ENERGY DETECTOR



B. PMC/BETA 201A VIBRATION METER

C. VITEC 650 VIBRATION METER



D. SKF CMVLIO PICOLOG VIBRATION / SEE / ACCELERATION ENVELOPE DETECTOR

E. SKF CMVP20 VIBRATION PEN

ILLUSTRATION A
EXAMPLES OF VARIOUS HAND-HELD OVERALL VIBRATION METERS

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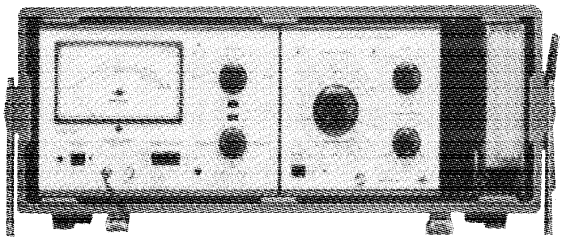
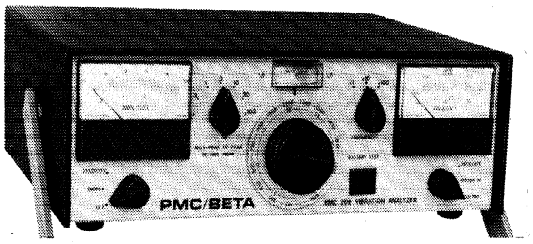
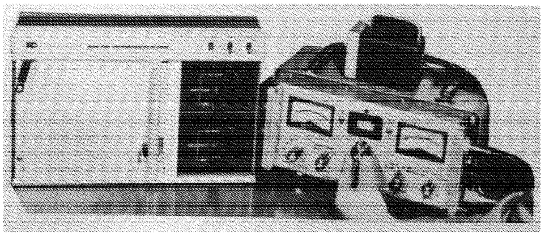


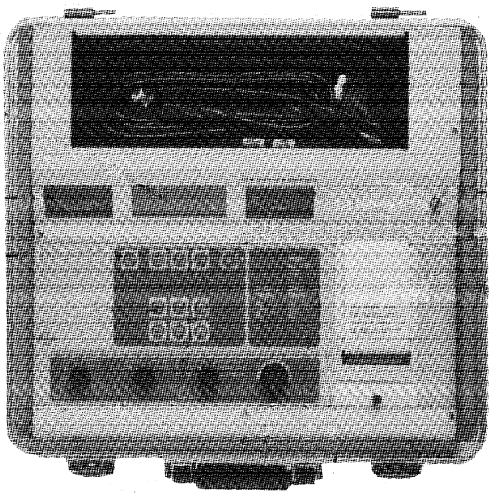
A. B&K 3513



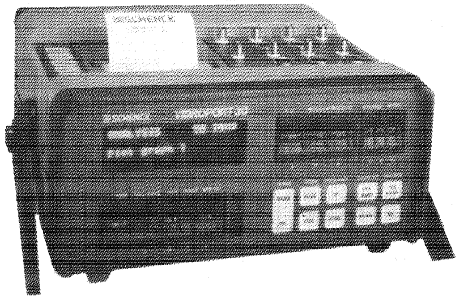
B. PMC/BETA 208



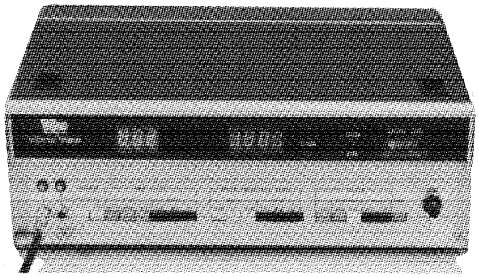
C. IRD 350 WITH
IRD 1080 PLOTTER



D. IRD 880



E. SCHENCK VIBRATION 30



F. VITEC VIBRA/VIEW

ILLUSTRATION B
EXAMPLES OF PORTABLE SWEPT-FILTER ANALYZERS

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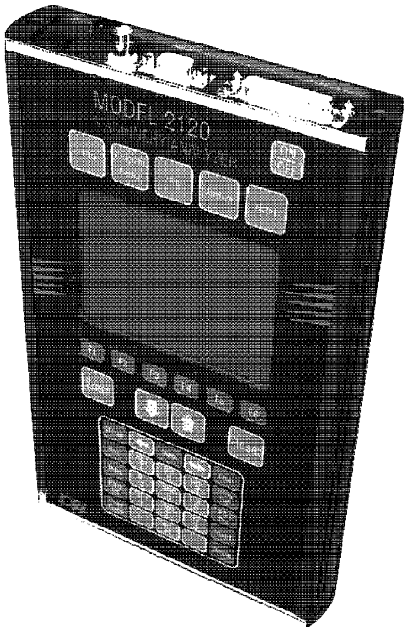
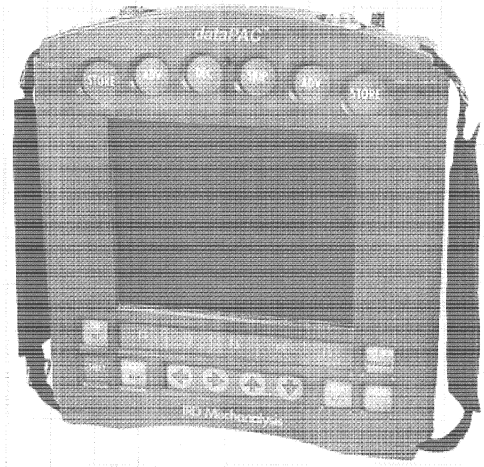
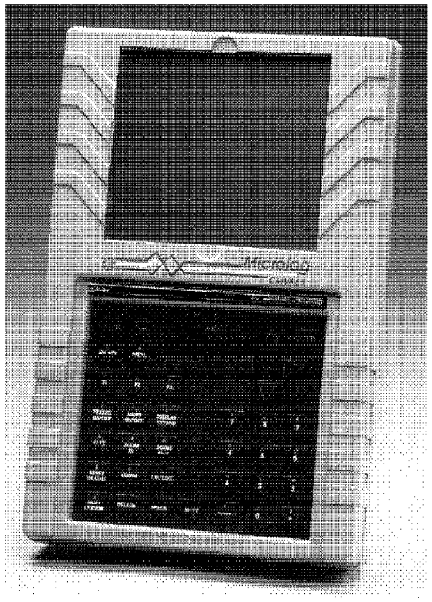


A. CSI 2120
MACHINERY ANALYZER



ENTEK IRD DataPac 1500



C. SKF MICROLOG CMVA55

D. PREDICT/DLI WATCHMAN DC-7B
(3 CHANNEL)

ILLUSTRATION C
EXAMPLES OF VARIOUS DATA COLLECTORS

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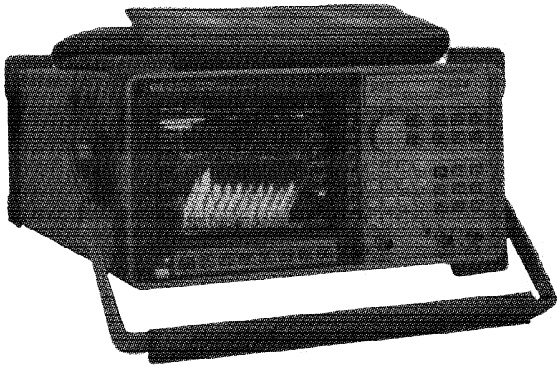
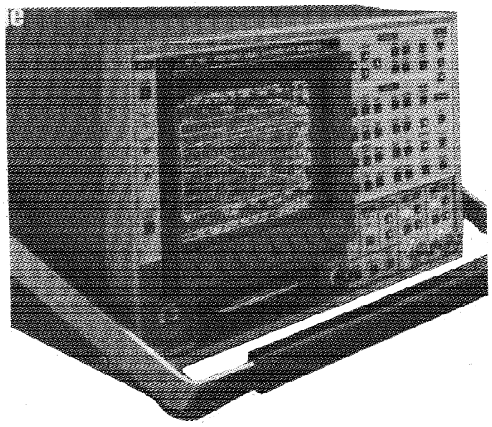
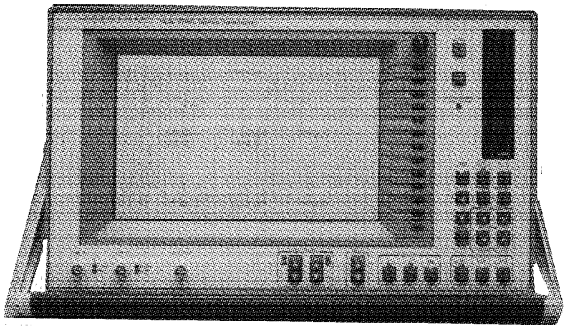


A. HEWLETT PACKARD 35670A
DYNAMIC SIGNAL ANALYZER
(2 OR 4 CHANNEL)



B. ONO SOKKI CF-360
(2 CHANNEL)



C. ROCKLAND 90665L
DYNAMIC SIGNAL ANALYZER
(2 CHANNEL)

D. SIGNAL PROCESSING SYSTEMS SA 390
DYNAMIC SIGNAL ANALYZER
(2, 4, 6, OR 8 CHANNEL)

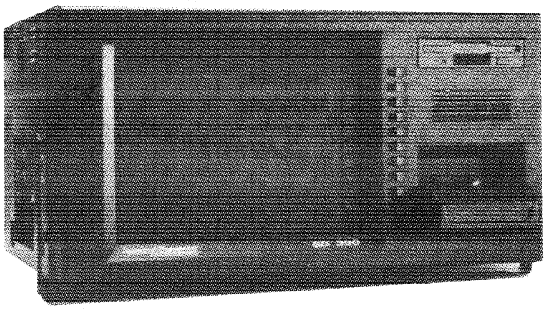


ILLUSTRATION D
EXAMPLES OF VARIOUS REAL-TIME ANALYZERS

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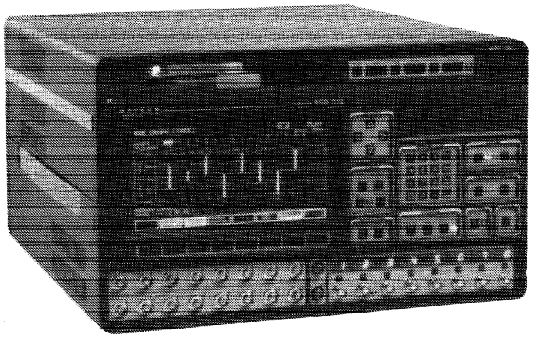
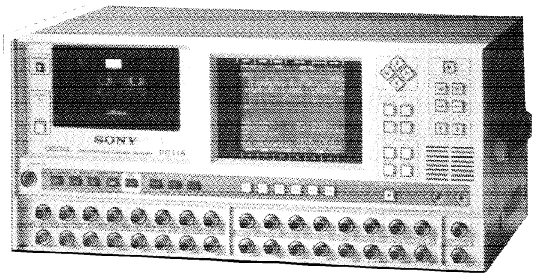
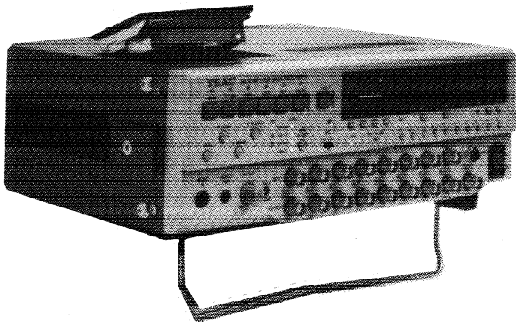


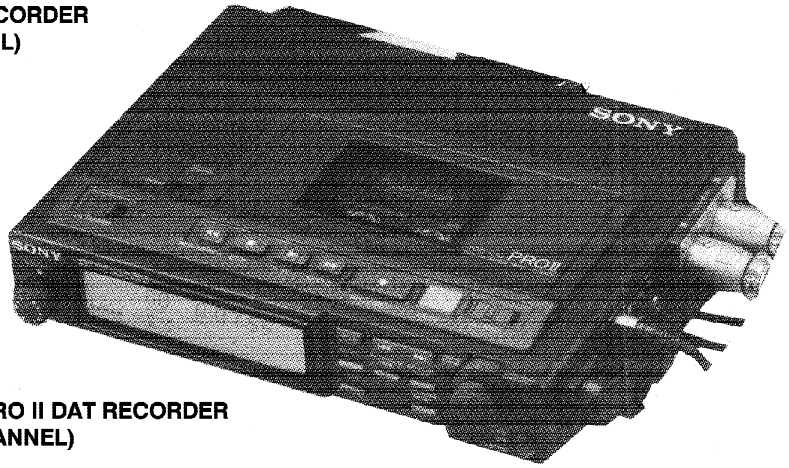
A. METRUM RSR 512 DAT RECORDER
(8,16,24 OR 32 CHANNEL)



C. SONY PC 116 DAT RECORDER
(2 THRU 16 CHANNEL)



B. TEAC RD-130T DAT RECORDER
(2, 4, OR 8 CHANNEL)



D. SONY TCD-D10 PRO II DAT RECORDER
(2 CHANNEL)

ILLUSTRATION E
EXAMPLES OF INSTRUMENT QUALITY TYPE RECORDERS

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2.3 OVERVIEW OF VIBRATION TRANSDUCERS AND HOW TO PROPERLY SELECT THEM

2.31 INTRODUCTION

Each of the following transducers will be covered:

1) **Accelerometers**
2) **Velocity Pickups**
3) **Non-contact Eddy Current Displacement Probes**
4) **Shaft Contact Displacement Probes (including Shaft Sticks and Shaft Riders)**

Figure 22 includes the three most common transducers in use today which include the accelerometer, velocity pickup and non-contact displacement probes. Table III is a general summary of the various categories of accelerometers, velocity pickups, and non-contact probes showing the more important characteristics and general specifications. In addition, model numbers of such transducers made by various manufacturers are included as examples.

In this chapter the optimum applications of each of these transducers (depending on the measurement to be made), mounting techniques and their influence on the accuracy of the vibration measurement will be discussed.

It is important to note that the following information is meant to be a general overview of transducer design and utilization. The specific design characteristics for each transducer may vary individually from vendor to vendor. Therefore, the reader should use this chapter as a guide for discussing his needs with a qualified vendor before purchasing a particular transducer or set of transducers.

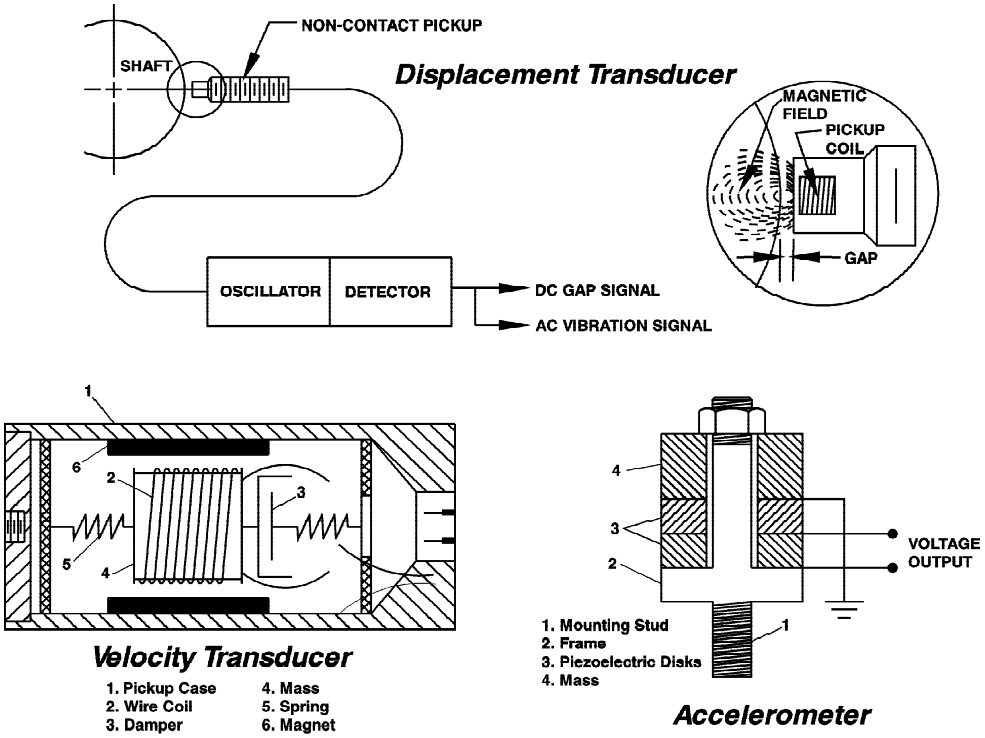

FIGURE 22
EXAMPLES OF VARIOUS TYPES OF VIBRATION TRANSDUCERS

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2.32 TYPES OF VIBRATION TRANSDUCERS AND THEIR OPTIMUM APPLICATIONS

It is important that a vibration analyst understand the limitations and optimum applications of each of these four types so that he can use them to his best advantage. At times, each type has certain characteristics that justifies its use in a particular application over the other types when monitoring machinery. *Accelerometers and velocity pickups placed on bearing housings measure absolute vibration while non-contact displacement probes measure relative vibration.*

2.321 Accelerometers

As the name implies, accelerometers are sensors which provide the direct measurement of acceleration (g). It is the piezoelectric element in the accelerometers that produces a signal proportional to acceleration. Accelerometers measure absolute vibration.

Accelerometers are the most common PMP transducers in use today due to 3 primary reasons: (1) they are relatively inexpensive when compared to velocity pickups; (2) *their frequency range capabilities are much wider* than those of velocity pickups, non-contact probes, shaft sticks, or shaft riders; and (3) *much more funding is being expended in research and development of a large variety of accelerometers*, not only to lower their cost, but also to enable them to accurately measure both lower and higher frequency data and to withstand harsh environmental conditions (high temperatures, operation in submerged oil baths or under water, exposure to corrosive gases or liquids, etc.).

There are three primary performance characteristics of accelerometers which affect their performance. These include voltage sensitivity (mV/g), frequency response (Hz or CPM), and mass (grams). In order to determine the acceleration in g's from the voltage generated by the piezoelectric crystal in the accelerometer, consider the following example:

EXAMPLE: If 2 Volts are generated by an accelerometer, which has a sensitivity of 100 mv/g, the acceleration would be:

$$A = \frac{2 \text{ Volts}}{.1 \text{ Volt/g}} = \underline{\underline{20g}}$$

In most cases, *if desiring to take low frequency measurements, it will be necessary to choose one of the low frequency accelerometers listed in Table III which typically are much higher in weight and normally have voltage sensitivities (mV/g) much higher than the general purpose accelerometers.* The reason for this high voltage sensitivity is to bring the vibration signal above the noise. Even though the displacement (mils) levels may be high, the acceleration levels (g) will be low for vibration at frequencies of less than 60 CPM (1 Hz). On the other hand, if desiring to take high frequency measurements (typically above 600,000 CPM or 10,000 Hz), it will likely be necessary to acquire one of the high frequency accelerometers listed in Table III. Typically, these high frequency accelerometers are much smaller with a lower voltage sensitivity, usually on the order of 10 mV/g or less.

One of the common misconceptions is that the higher the voltage sensitivity (mV/g), the better the accelerometer. This is not always the case. For example, some seismic accelerometers used in low frequency measurements are very sensitive to temperature changes and, if dropped, can fail due to the instantaneous voltage surge which oversaturates their built-in electronics. Conversely, a general purpose or high frequency accelerometer can be dropped with no damage to the transducer.

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TABLE III
GENERAL TRANSDUCER CHARACTERISTICS

TYPICAL SPECIFICATIONS	ACCELEROMETERS					VELOCITY PICKUPS		NONCONTACT DISPLACEMENT PROBES
	GENERAL PURPOSE	LOW FREQUENCY	HIGH FREQUENCY	PERMANENTLY MOUNTED	TRIAXIAL	SEISMIC	PIEZOELECTRIC	
TYPICAL SENSITIVITY RANGE (mV/g Unless Noted)	10 – 100	500 – 10,000	0.4 – 20	10 – 100	10 – 100	(mV/in/s) 100 – 1000	(mV/in/s) 100 – 1000	(mV/mil) 100 – 200
TYPICAL MEASUREMENT FREQUENCY RANGE (CPM)	60 – 600,000	6 – 60,000	600 – 3,600,000	180 – 600,000	120 – 600,000	600 – 60,000	60 – 270,000	0 – 120,000
TYPICAL STUD MOUNTED NATURAL FREQUENCIES (CPM)	960,000 – 2,700,000	390,000 – 2,100,000	4,200,000 – 10,800,000	1,080,000 – 1,800,000	480,000 – 1,200,000	APPROXIMATELY 200,000	APPROXIMATELY 200,000	N.A.
TYPICAL WEIGHT RANGE (grams)	17 – 110	135 – 1000	1.2 – 15	60 – 180	17 – 130	100 – 150	150 – 250	230 – 280
TYPICAL USABLE TEMPERATURE RANGE (°F)	(–100°) – 250°	(–100°) – 250°	(–40°) – 350°	(–50°) – 250°	(–100°) – 250°	(–50°) – 250°	(–50°) – 250°	(–30°) – 350°
EXAMPLE TRANSDUCER MODEL NUMBERS AND MANUFACTURER (Alphabetical Order)	B & K 4370/81 B & K 4390 IRD 943 IRD 970 PCB 302A06 PCB 302A09 PCB 307A PCB 308B RION PV–44A VIBRAMET. 1020 VIBRAMET. 1021 WILCOXON 793	B & K 8318 (S) PCB 326A01 (S) PCB 326A02 (S) PCB 326A03 (S) PCB 393C (C) VIBRAMET. 5100 (S) WILCOXON 731 (C) WILCOXON 793L (C) WILCOXON 797L (S) (S)–SHEAR MODE (NOT TEMPERATURE SENSITIVE; BRIEF SETTLING TIME) (C)–COMPRESSION MODE (TEMPERATURE SENSITIVE; LONG SETTLING TIME)	B & K 8309 PCB 303A02 RION PV 90B VIBRAMET. 6032 WILCOXON 732	B & K 8310/24 B & K 8317 IRD 941B IRD 942 PCB 328B VIBRAMET. 1024	B & K 4321 PCB 306A PCB 306A06 VIBRAMET. 3130 WILCOXON 733	BENTLY. 47633 IRD 544	BENTLY. 26862 IRD 560 WILCOXON 756V WILCOXON 793V WILCOXON 797V	BENTLY. 3300 BENTLY. 7200 IRD 403

NOTES:

1. For detailed information on particular transducers, see the APPENDIX which includes specifications for many of them.
2. By inclusion of specific transducers within this table, Technical Associates does not necessarily recommend them.

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Note that Table III also lists some accelerometers that are specially designed for permanently mounting even in harsh environments. Sometimes they are placed under water or in lubricating or cutting oils. *Others are designed to make triaxial measurements* (simultaneous measurements in the horizontal, vertical and axial directions) to increase the measurement speed on a PMP route.

Figures 23 thru 25 illustrate the three most commonly used piezoelectric accelerometers. There are two major types. Figures 23 and 24 show one type, called the *"compression mode accelerometers".* Note that the accelerometer in Figure 24 is an inverted compression type. Figure 25 shows the other major type, known as a *"shear mode accelerometer"*. Until recently, the compression designs in Figures 23 and 24 probably have been the most widely utilized designs, mainly due to their simplicity and lower cost, along with their wide frequency range.

However, one of the disadvantages of compression mode accelerometers is that they are often adversely affected by thermal transients and base strain sensitivity which can oversaturate their electronics. *The time it will require for the transducer to "settle" will be much longer than that expended for a shear mode accelerometer (discussed below), particularly when making low frequency measurements. In those situations where an accelerometer will be subjected to large differences in temperature* (either due to a temperature change in the mounting surface or high pressure air blowing continuously on the accelerometer), *the shear type accelerometer pictured in Figure 25 may be a better choice* since the crystal element is isolated from the base and housing by being sandwiched between the seismic mass and a center post. *The*

advantages of shear mode accelerometers are a stable output signal (especially when measuring at low frequencies) and smaller size and mass. The disadvantage of these units is the higher cost (in most cases) due to the added components required to make up the shear configuration.

This difference between shear mode and compression mode accelerometers is particularly evident when an analyst is taking low frequency measurements (particularly below 60 CPM or 1 Hz). In this case, *a compression mode accelerometer may require 3 or 4 minutes to stabilize before any measurements can be taken whatsoever.* However, *if a shear mode accelerometer was used instead, it will not be nearly so sensitive to thermal transients.* It will stabilize quickly, allowing the analyst to begin his measurement almost immediately, or certainly within 20 seconds or so.

Another important difference between piezoelectric accelerometers are *the two types of signal conditioning electronics* that they can have. Each of these is pictured in Figure 26: (1) *A high impedance, charge mode type* requires an external signal conditioner; and (2) *a low impedance, voltage mode, "ICP" type* contains built-in signal conditioning electronics and does not normally require any external signal conditioning (meaning that the accelerometer can be powered directly from ICP[15] circuitry built right into most data collectors and real time analyzers today. Each of the three accelerometers pictured in Figures 23 thru 25 are the "ICP" (integrated circuit piezoelectric) type - the most commonly used accelerometer types in programs today.

In the case of the charge mode unit, its accelerometer sensitivity is usually defined in units of picocoulombs per g (pC/g), whereas the sensitivity of voltage-mode units is normally expressed directly in millivolts per g (mV/g). These charge mode accelerometers will most always require an external charge amplifier/signal conditioner in the field to use them.

The built-in microelectronics conditions the signal from the crystal within the ICP accelerometer to a low impedance voltage which is compatible with the readout device. To turn on and perform its measurement, a constant current power source (normally available in most all FFT analyzers today) is required for the transistor within the accelerometer. *The advantages of the ICP accelerometers are: (1) their fixed sensitivity; (2) the ability to operate in dirty and moist environments reliably; (3) they only need an ordinary 2-wire coaxial cable which can be usually powered without additional power sources up to about 1000 feet; (4) they do not need a separate signal conditioner when one is available in the analyzer; and (5) the system usually costs less. One disadvantage is a limited temperature range due to the survivability of the built-in electronics* (although this can sometimes be overcome by use of special ceramic construction materials). Another disadvantage occurs if the accelerometer is not hermetically sealed - *its low frequency performance will be degraded significantly if moisture enters into the transducer.* Reference 4 points out that "all accelerometers breathe, unless they are certified as hermetically sealed, and during this micro-breathing, moisture inevitably penetrates." *Therefore, it is important to acquire a hermetically sealed unit, particularly if it is to be used in humid environments, and especially in those cases where low-frequency measurements are to be conducted.*

On the other hand, one advantage of the charge mode accelerometer is the ability to operate at high temperatures. However, one of the biggest disadvantages of the charge mode accelerometer is that a separate charge amplifier is required between the analyzer and the sensor, as shown in Figure 26. Because of this disadvantage and other disadvantages, such as requiring a special low-noise, fixed length cable and a clean and moisture-free environment, they are *not used in most predictive maintenance measurement systems today.*

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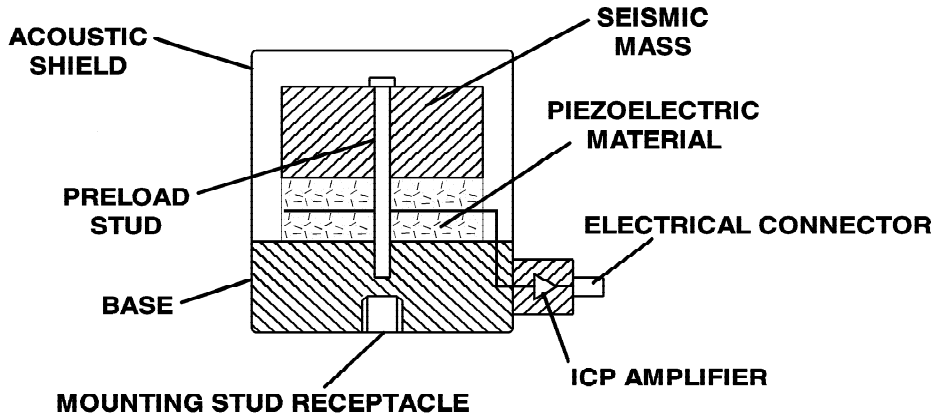

FIGURE 23
COMPRESSION TYPE ACCELEROMETER (Ref. 3)

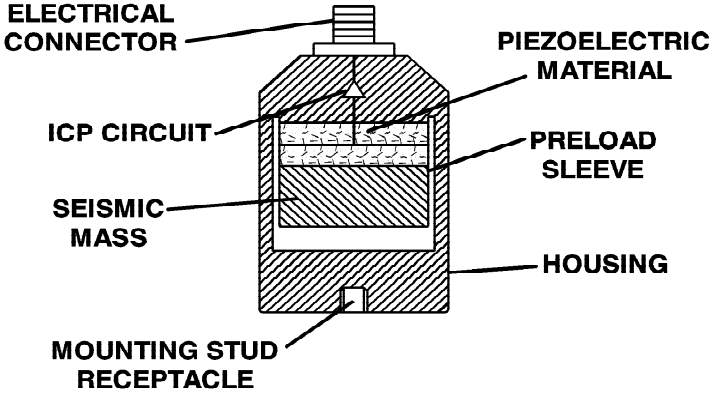

FIGURE 24
INVERTED COMPRESSION TYPE ACCELEROMETER (Ref. 3)

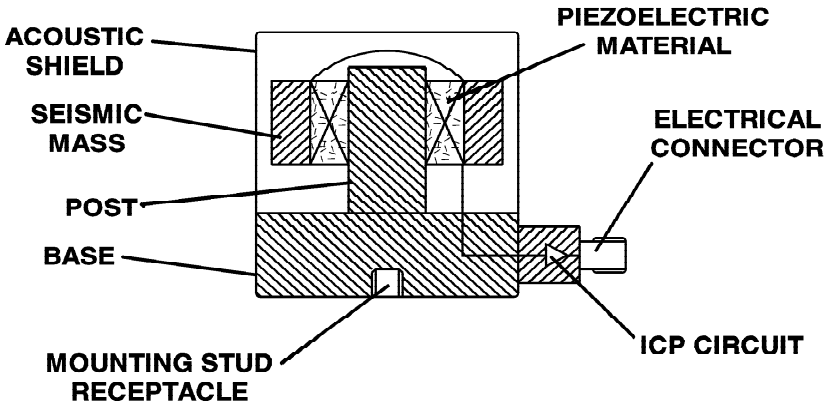

FIGURE 25
SHEAR TYPE ACCELEROMETER (Ref. 3)

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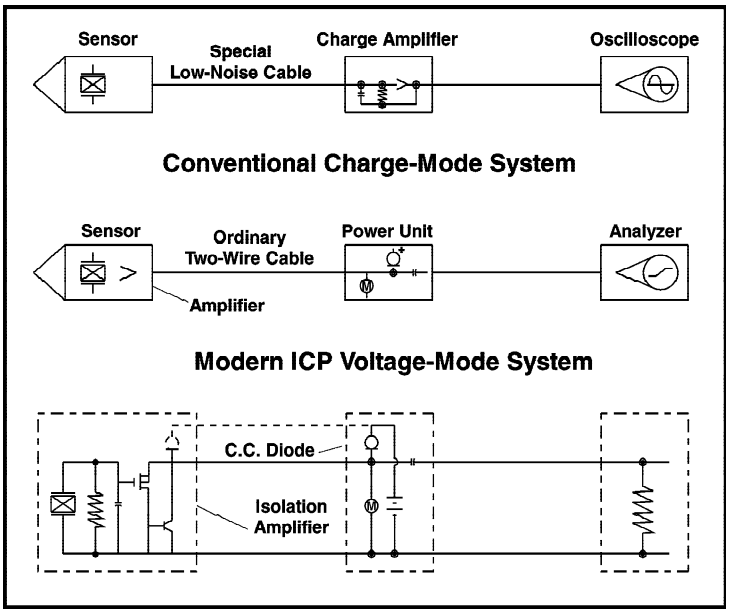

FIGURE 26
COMPARISON OF CHARGE MODE AND ICP VOLTAGE-MODE ACCELEROMETERS (Ref. 4)

When choosing an accelerometer, it is important to be aware of the resonant frequency and phase response of each accelerometer. This is illustrated by Figure 27 which shows an accelerometer with a fairly constant phase response from approximately 5% up to about 60% of its mounted natural frequency (f_m). *Note that as the mounted natural frequency of an accelerometer is approached, a dramatic phase change takes place.* The phase shifts almost 180° after passing through resonance. Therefore, when an accelerometer is used to perform a phase analysis on a machine, the frequency range at which this phase response occurs must be taken into consideration, particularly on very high speed machinery. If not, the accuracy of the phase analysis could be adversely effected. Furthermore, *if a magnet or an extension probe is used to mount an accelerometer, the natural frequency of the system will be considerably lower than the accelerometer stud mounted. As a result, the frequency ranges from which accurate phase measurements can be obtained will be considerably lower.*

Accelerometers are available with widely different frequency responses. For Example, *frequency ranges go from as low as .01 Hz (.6 CPM) up to over 60,000 Hz (3,600,000 CPM); voltage sensitivities will vary from about 0.4 mV/g up to 10,000 mV/g (10 V/g); accelerometer masses can vary from 1 gram up to over 1000 grams*. Since most predictive maintenance programs include not only normal speed machinery (600 to 3600 RPM), but also low-speed (particularly below 200 RPM), as well as high-speed machinery (particularly above 10,000 RPM), *it will be necessary (in most cases) to have at least three different accelerometers - a low frequency, a general purpose, and a high frequency accelerometer. Most likely, the general purpose accelerometer can be used successfully to make over 90% of the measurements*. However, without the low and high frequency accelerometers, the accuracy of the measurements from the very low or very high speed machines will be reduced. Therefore, the data integrity will be compromised.

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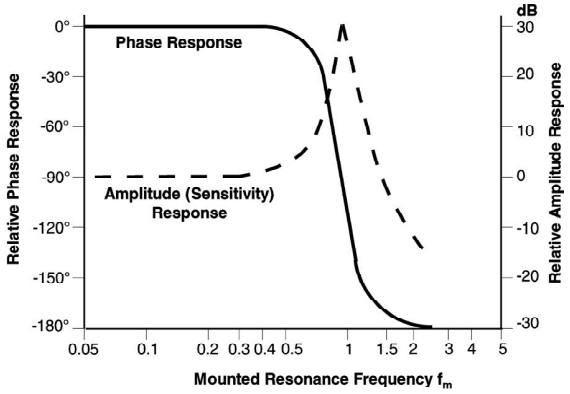

FIGURE 27
BODE' PLOT OF ACCELEROMETER SENSITIVITY AND PHASE RESPONSE AS A FUNCTION OF FREQUENCY (Ref. 4)

2.322 Velocity Pickups

For most applications today, velocity pickups have been replaced by accelerometers because of their limited frequency ranges and relatively higher cost. Also, these units are typically very large in size and weight (the example in Figure 28 weighs 21 ounces or 595 grams), which can undesirably contribute to the mass load of the machine or structure being evaluated. However, *one advantage of a velocity pickup is that it does not have to be powered by an external power supply* (such as an ICP current or a separate charge amplifier). Also, since velocity is the commonly preferred unit of vibration measurement in most PMP programs, *the signal does not have to be integrated from acceleration to velocity (as is the case with accelerometers).* Therefore, the integration noise which often appears in the first 2 to 4 lines of the FFT spectra taken by accelerometers integrated to velocity does not occur with these pickups (however, the velocity pickups can still produce noise within the first few lines of the spectrum due to their electronics, the insufficient analyzer input sensitivity and depending on the frequency span chosen).

There are two types of velocity transducers - moving coil seismic velocity transducers and piezoelectric velocity pickups. A seismic velocity transducer is shown in Figure 28 while example piezoelectric velocity pickups are shown in Figures 29 and 30. Velocity transducers also measure absolute vibration, like accelerometers.

Unlike piezoelectric transducers (piezoelectric accelerometers and piezo-velocity), *the moving coil seismic velocity pickup does not require an external power supply of any kind. The voltage is self-generated.* The seismic mass and coil remain stationary since they are spring-suspended, whereas the permanent magnet surrounding the coil will oscillate since it is firmly attached to the transducer case. A voltage (proportional to the relative velocity of the magnet and coil) is generated and then transmitted from the transducer into a vibration analyzer.

Since its seismic mass is supported by soft springs, this transducer has a very low natural frequency of about 10 - 20 Hz (600 - 1200 CPM). The suspended mass shown on Figure 28 is damped (either electrically or with synthetic oil). Therefore, *the transducer's low frequency response is limited by its first natural frequency.* As the mass overcomes the damping effect in low frequencies and begins to move in phase with the vibration, its sensitivity will drop because the coil will hardly be cutting through the magnetic flux. For example, *if an IRD 544 velocity pickup like that shown in Figure 28 was used to measure vibration below 10 Hz (600 CPM), a Correction Factor Chart, as shown in Figure 29, would have to be used.* If a measurement of .1 in/sec at 300 CPM were recorded, Figure 29 shows that the actual level

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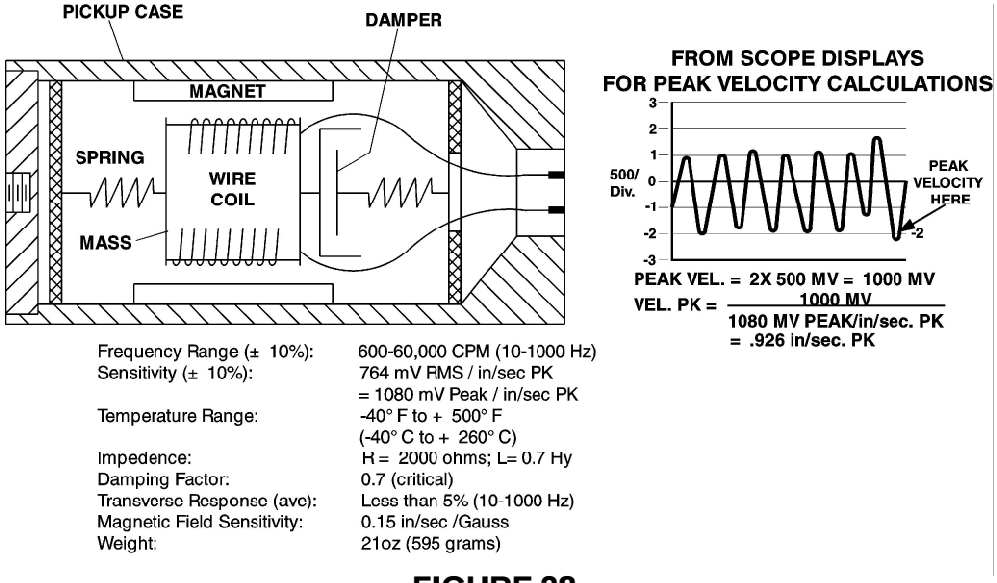

Frequency Range (± 10%):	600-60,000 CPM (10-1000 Hz)
Sensitivity (± 10%):	764 mV RMS / in/sec PK = 1080 mV Peak / in/sec PK
Temperature Range:	-40° F to + 500° F (-40° C to + 260° C)
Impedence:	H = 2000 ohms; L= 0.7 Hy
Damping Factor:	0.7 (critical)
Transverse Response (ave):	Less than 5% (10-1000 Hz)
Magnetic Field Sensitivity:	0.15 in/sec /Gauss
Weight:	21oz (595 grams)

FIGURE 28
IRD 544 SEISMIC VELOCITY PICKUP (Ref. 5)

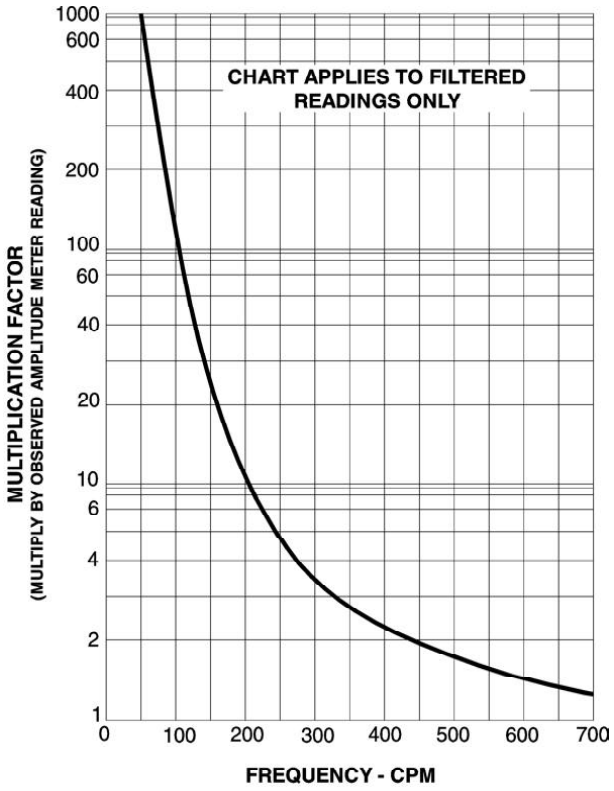

FIGURE 29
CORRECTION FACTOR CHART FOR IRD 544 SEISMIC VELOCITY PICKUP WHEN MEASURING FREQUENCIES BELOW 600 CPM (10 Hz) (Ref. 5)

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would be about 3.3 times higher (or about .33 in/sec). Note that *with decreasing frequencies, significantly higher and higher multiplication factors must be applied to correct the vibration readings. Therefore, a seismic velocity pickup is not a good low frequency transducer.*

Another problem with most seismic velocity pickups is the upper frequency response which is commonly limited to no more than about 1000 - 2000 Hz (60,000 to 120,000 CPM). *If a much lower or higher frequency measurement is required, a piezoelectric velocity transducer, as shown in Figures 30 or 31, could be used, particularly if a direct velocity reading is preferred. The direct velocity reading is provided by an internal analog integrator inside the transducer which converts the acceleration signal to velocity.* The noise which generally appears at low frequencies due to integration and other sources is suppressed by having this analog integration performed within the transducer rather than by an integration circuit in the instrument. In comparison to seismic velocity transducers, *the frequency range of piezoelectric velocity pickups is much wider.* For example, note that the unit in Figure 30 has a (±10%) frequency response of 60 - 270,000 CPM (1 - 4500 Hz) and that of Figure 31 is from 150 - 210,000 CPM (2.5 - 3500 Hz); while the seismic unit in Figure 28 has a (±10%) frequency response of only 600 - 60,000 CPM (10 - 1000 Hz).

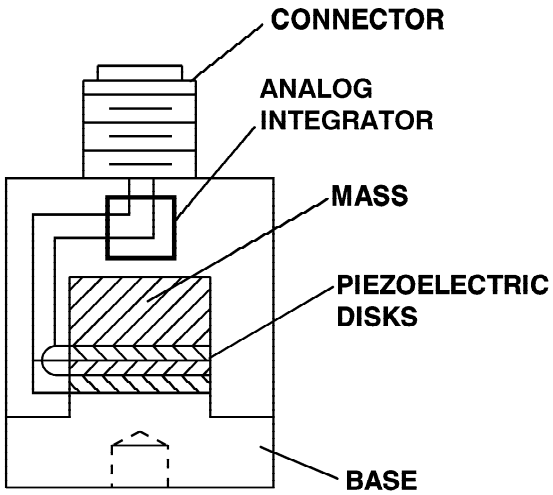

FIGURE 30
IRD 560 PIEZOELECTRIC VELOCITY PICKUP (Ref. 5)

Frequency Range (± 10%):	60-270,000 CPM (1-4500 Hz)
Sensitivity (± 10%):	764 mV RMS / in/sec PK
	= 1080 mV Peak / in/sec PK
Temperature Range:	-40° F to + 250° F
	(-40° C to + 121° C)
Mounted Natural Freq.:	600,000 CPM min. (10,000 Hz)
Transverse Response :	5% max @ 600 CPM (10Hz), 5 g peak
Magnetic Field Sensitivity:	.00001 in/sec /Gauss
Weight:	38 oz (1079 grams)

Figure 30 shows a piezoelectric disk mounted below the seismic mass. Be aware that this disk introduces only a small amount of phase shift. This makes *a piezoelectric velocity pickup a much better choice for phase analysis and balancing than a seismic velocity pickup.* In addition, unlike seismic transducers, *piezoelectric velocity pickups and accelerometers are virtually unaffected by magnetic fields* (see comparative magnetic field sensitivities in Figures 28 and 30, respectively), *making them ideal for analyzing electric motors.*

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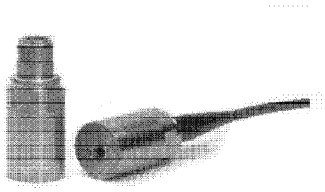
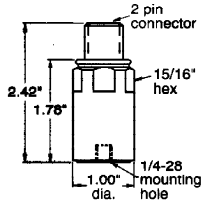
WILCOXON RESEARCH

INDUSTRIAL

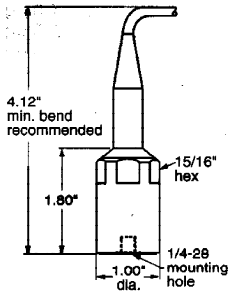


793V Series
Piezoelectric Velocity Transducers

FEATURES:
- Intrinsic safety certified option
- Industrial ruggedness
- Eliminates distortion caused by high frequency signals
- Corrosion-resistant
- Internally integrated to velocity
- Ultra low-noise electronics for clear signals at very low vibration levels
- Miswiring protection



793V Series Models



793VS Series Models

SPECIFICATIONS	All Models	793V	793V-5	Units
DYNAMIC				
Sensitivity, ±10%, 25°C	—	100	500	mV/in/sec
Velocity Range	—	50	10	in/sec peak
Amplitude Nonlinearity	1%			
Frequency Response:				
±10%	—	2.0 - 3,500	8.0 - 3,500	Hz
±3 dB	—	1.5 - 7,000	5.0 - 7,000	Hz
Resonance Frequency, mounted, nom.	15 kHz			
Transverse Sensitivity, max.	5 % of axial			
Temperature Response	see graph			
ELECTRICAL				
Power Requirement:				
voltage source	18 - 30 VDC			
current regulating diode[1]	2 - 10 mA			
Electrical Noise, equiv. in/sec, nominal:				
Broadband 2.5 Hz to 25 kHz	—	100	40	µin/sec
Spectral 10 Hz	—	10	4.0	µin/sec/√Hz
100 Hz	—	1.0	0.4	µin/sec/√Hz
1000 Hz	—	0.2	0.05	µin/sec/√Hz
Absolute Phase Shift, nom. the greater of:	—	$\tan^{-1}$ 2/f or 2°	$\tan^{-1}$ 5/f or 2°	
Output Impedance, nominal 4mA supply,				
the greater of:	—	5,000/f	25,000/f	
	—	or 200Ω	or 200Ω	
Bias Output Voltage, nominal	10 VDC			
Grounding	case isolated, internally shielded			
ENVIRONMENTAL				
Temperature Range	–50 to 120°C			
Vibration Limit	250 g peak			
Shock Limit	2,500 g peak			
Electromagnetic Sensitivity, equiv. in/sec	25 µin/sec/gauss			
Sealing	hermetic			
Base Strain Sensitivity	0.0005 in/sec/µstrain			
PHYSICAL				
Weight	145 grams			
Case Material	316L stainless steel			
Mounting	¼ - 28 tapped hole			
Output Connector	MIL-C-5015 2-pin			
Connections Pin A ..... signal, power				
Pin B ..... common				
Cabling[2]: Mating Connector	Wilcoxon R6 (Amphenol 97-3106A-10SL-4S)			
Recommended Cable	J9T2, two conductor shielded, Tefzel jacket, 30 pF/ft.			

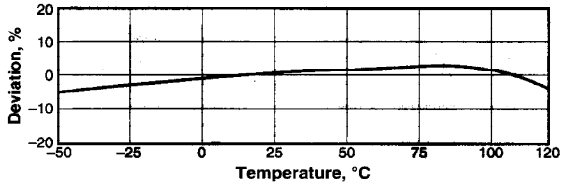
NOTES: [1] A maximum current of 6 mA is recommended for operating temperatures in excess of 100 °C.
[2] Cabling for "S" option model is J61, coaxial, polyurethane jacket, 30 pF/ft.

OPTIONS: Intrinsic safety certification; stainless steel flexible hose; 10 ft. sealed integral cable "S" or "SG", temperature sensor "T", customer specified sensitivity, low frequency filtering, cable length, connector.

ACCESSORIES SUPPLIED: SF6 mounting stud, calibration data, CCD card

ACCESSORIES AVAILABLE: R6SL-J9T2-XX splash-proof cable assembly, magnetic mounting bases, SF8 cementing studs, power supplies, amplifiers, signal conditioners.

TYPICAL TEMPERATURE RESPONSE



TYPICAL FREQUENCY RESPONSE

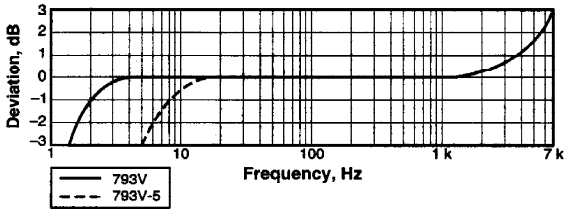


FIGURE 31
793V SERIES PIEZOELECTRIC VELOCITY TRANSDUCERS (Ref. 5)

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2.323 Non-contact Eddy Current Displacement Probes

Non-contact eddy current displacement probe systems like that shown in Figures 32 and 33 (sometimes called proximity probes) are *used primarily to measure relative shaft vibration, radial and axial shaft position, and the differential expansion between the case and the rotor.* Accelerometers and velocity pickups measure *absolute* vibration. If the overall peak velocity is measured on a bearing cap, this is a measure of its motion. Non-contact pickups are *especially effective on large machinery outfitted with oil film plain bearings such as turbine/ generators, compressors, large motors, etc.* Thus, when ordering such large, high-speed machinery, a plant may wish to include in the specifications non-contact displacement probes that are to be internally mounted close to the shaft in the heavy casing. *In this case, it is expedient to measure the vibration directly at the shaft since much of the energy will not transmit through the heavy casings.* By comparison, if it was suggested to measure the vibration on the exterior surface of the housing (as it is done typically by PMP personnel on a data collection route) with an accelerometer or velocity pickup magnetically mounted sensing absolute motion), the vibration energy originating from the shaft can be greatly attenuated by the time it is sensed by the bearing cap transducer. Thus, *a non-contacting displacement transducer would be a better selection for these machines, particularly for measurements at operating speed and up through the first five or six harmonics of operating speed.*

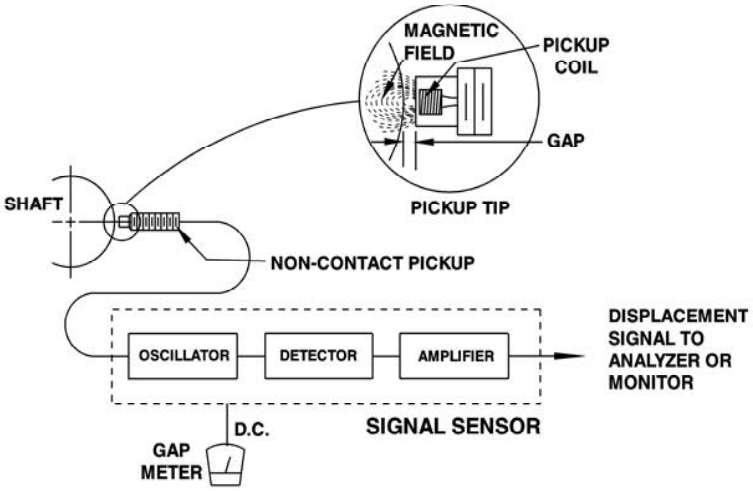

FIGURE 32
SCHEMATIC DIAGRAM OF AN EDDY CURRENT NON-CONTACT DISPLACEMENT PROBE SYSTEM (Ref. 5)

Be aware that the readings from a non-contact eddy current displacement probe (measuring relative shaft vibration) *may be quite a bit higher in amplitude than those taken on the casing itself. In the case of a journal bearing machine where the shaft is 3 to 6 feet from the outside, the shaft vibration may be as much as 20 times higher than the reading obtained on the bearing housing.* The signal is attenuated as it passes through the oil film, as well as through each of the metal interfaces and thicknesses out to the machine housing. The amount of signal loss depends on the oil-film characteristics, the mass, and the distance from the shaft out to the bearing housing surfaces. *This means that potentially serious problems, such as oil whirl or oil whip, may be missed altogether if readings are not taken directly from the shaft itself.* Therefore, *the best possible vibration spectrum, particularly for events at 1X RPM up through about 6X RPM, is that which can be acquired directly from a non-contact eddy current probe . This non-contact probe will measure the vibration of the shaft relative to the mounting system of the probe itself.* Therefore, it is most important to anchor the probe as much as possible to get the truest picture of shaft vibration.

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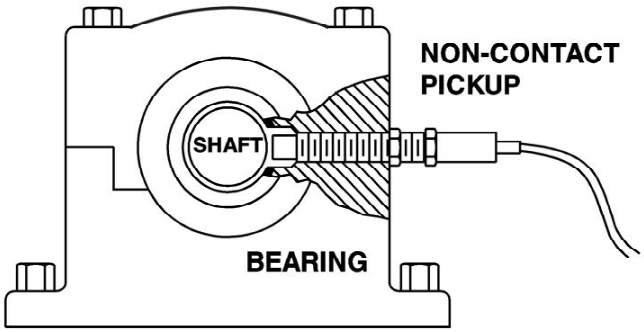
Unlike piezoelectric accelerometers and velocity pickups, a non-contact sensor requires external electronic circuitry to generate a very high frequency AC signal since it does not have an internal component which generates a voltage or an electrical charge in response to the vibration. As shown in Figure 32, this high frequency electrical signal is applied through a coaxial cable to the coil which generates a magnetic field at the tip of the pickup. Then the tip will be placed at a gap of only about 40 to 60 mils from the target surface. The shaft will absorb some of the magnetic energy. As the shaft moves relative to the pickup tip, *the probe will provide a peak-to-peak AC signal voltage proportional to the vibration, and a DC signal proportional to the gap. The oscillations in the AC signal caused by vibration of the shaft are detected and can be read by a permanent monitor or an analyzer connected to an output of the non-contact probe to determine the amount of relative shaft vibration while the DC signal can be used to monitor the change in gap.* A non-contact probe can be particularly useful if it is targeted on the shaft's end *to detect axial shaft movement and possible wear of thrust bearings.*

Figure 33 shows a non-contact probe mounted in a bearing housing. The signal from the probe can be taken directly into an analyzer for a measurement. Be very careful! *Do not connect the cable to the ICP port on a spectrum analyzer or data collector. The ICP current will cause the vibration "readout" to suddenly jump which can cause these machines to alarm, or even worse, trip it off line.*



SHAFT VIBRATION FROM A 200 MV/MIL PROX. PROBE

1.0V AC = 1/.2 = 5.0 MILS
0.8V AC = .8/.2 = 4.0 MILS
0.5V AC = .5/.2 = 2.5 MILS
0.1V AC = .1/.2 = 0.5 MILS

NOTE: Above vibration levels are Peak-to-Peak relative vibration levels.

FIGURE 33
NON-CONTACT EDDY CURRENT PROBE MOUNTED IN A BEARING HOUSING (Ref. 5)

Figure 34 shows a typical response curve for a non-contact probe. *This curve represents the change in sensor output voltage as the distance between the tip of the pickup and the target material is altered.* Note that the relationship of the output voltage to the gap approximates a straight line. *The slope of this line in millivolts per mil (mV/mil) determines the non-contact probe sensitivity* (normally non-contact probe sensitivities range from about 100 to 200 mV/mil, peak-to-peak). The system shown in Figure 34 has a linear gap range from about 20 to 100 mils or a total of 80 mils. Therefore, *the optimum "setpoint" would be selected at the midpoint* (60 mils in this case) of the specified linear gap range. Note that most larger plain bearing machinery (turbines, boiler feed pumps, etc.) are outfitted with "X-Y" probes (with each probe normally located 45° off of vertical). This will be utilized to capture Lissajous Orbits which track actual shaft motion (to be covered in later chapters).

Of great importance when installing non-contact probes is to provide a very smooth, uniform surface finish for the target area. The American Petroleum Institute Standards (API 670) call for a surface finish of 16 - 32 microinches RMS (or 0.40 - 0.80 micrometers RMS). *The target area must be free of any surface imperfections such as scratches or high spots.* In fact, a common problem with non-contact probes is that, when a scratch is present, *the probe cannot discern between true vibration and the scratch depth, thereby causing an error in the readout.*

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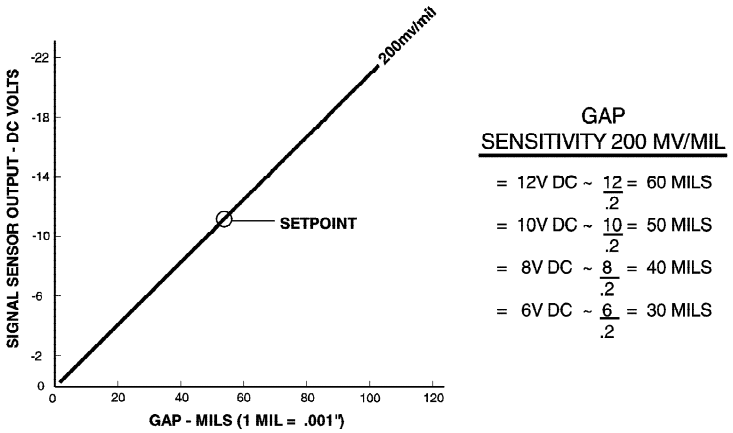

FIGURE 34
TYPICAL OUTPUT FOR AN EDDY CURRENT NON-CONTACT DISPLACEMENT PROBE SYSTEM (Ref. 5)

Another problem common to non-contact probe systems is either mechanical or electrical runout. Mechanical runout is an error which is caused by out-of-roundness and surface imperfections (scratches, chain marks, dents, rust or other conductive buildup on the shaft, flat spots and engravings). *It is most important to ensure that there is less mechanical runout in the shaft than the minimum allowable vibration displacement which is to be measured.* Readings should be taken at 30° - 45° rotational intervals around the shaft, plotting runout versus angular position to ensure this, and also to provide a baseline for comparison with electrical runout data. Figure 35 shows a waveform captured from a non-contact probe which was targeted on a shaft having a scratch in the target area. Note the pronounced pulses in the time domain spaced at once/revolution each time the scratch passed the probe.

Electrical runout is a signal error which occurs in eddy current displacement measurements when the shaft surface conductivity varies. That is, the target's surface itself has nonuniform electrical conductivity/resistivity/permeability properties. Another source of electrical runout is due to the presence of a localized magnetic field(s) on the shaft surface. This error will repeat exactly with each shaft revolution. Such shaft magnetism may result from magnetic-particle testing; or by operating in a strong magnetic field produced by motors, generators or alternators; or can even be induced from welding operations nearby. *In these instances,*

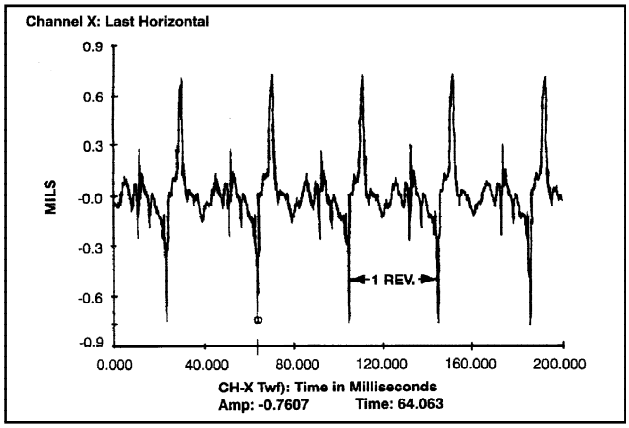

FIGURE 35
TIME WAVEFORM TAKEN FROM A NON-CONTACT PROBE TARGETED ON A SCRATCHED SHAFT (Ref. 10)

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it is impossible for the displacement probe to differentiate true vibration from either the mechanical or the electrical runout. Depending on where the electrical and mechanical runouts are located on the shaft, they may either add to or subtract from each other. That is, ***the displacement readout will be a vectorial addition of electrical runout, mechanical runout, and the actual vibration***. There are procedures in place for determining the presence of mechanical and electrical runout and for minimizing their effect. In the case of electrical runout, it often is necessary to correct this with a degaussing instrument.

Remember that *it is very important to calibrate the non-contact probe for the specific shaft material to which it will be targeted*. Figure 36 shows typical response curves for different target materials in comparison to the gap that is set between the probe and the target. This chart is used to set up and calibrate the response of the proximity probe. It is important to notice that "roll-off" problems occur as the gap is increased. *It is recommended that probes not be set up in these roll-off ranges.*

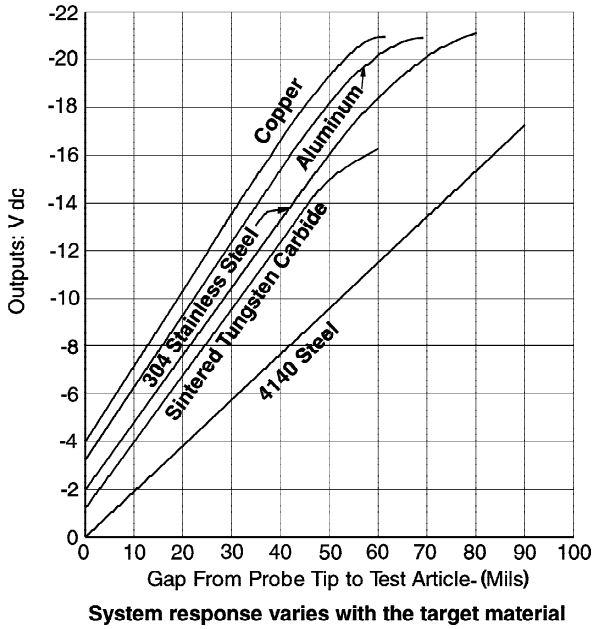

FIGURE 36
VOLTAGE RESPONSE VERSUS GAP FROM PROBE TIP TO SHAFT FOR VARIOUS TARGET MATERIALS (Ref. 8)

Finally, *a remaining problem with a non-contact probe is in the high frequency range.* Although specifications for non-contact probe systems will claim a frequency response up to 600,000 CPM, this does not actually occur in the real world. *Since the probe is measuring displacement, it will tend to emphasize low frequency sources, but de-emphasize high frequency sources.* For example, consider a centrifugal compressor outfitted with non-contact probes on each of its impellers which range in operating speeds from about 20,000 up to approximately 50,000 RPM. *Since the displacement amplitudes become very low with increasing frequencies, these displacement measuring systems will not be sufficiently sensitive to pick up problems with gear wear or with failing thrust bearings on these machines* (which are commonly outfitted with about 6 pads on each thrust bearing). However, if problems such as unbalance (due to impeller buildup and/or erosion) or misalignment occur, the system will readily respond, whether it is targeting a low-speed or a high-speed impeller. Still, however, such machines *should always be* outfitted with non-contact probe permanent vibration monitoring systems in order to detect the actual shaft vibration and the onset of problems to which they are in fact sensitive.

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2.324 Shaft Contact Displacement Probes

Shaft contact probes (shaft sticks and shaft riders) are probes that directly measure shaft vibration by actually riding on the shaft surface. Although they have been replaced by non-contact eddy current probes in most cases today, they are still used in cases where a proximity probe has not been installed, for balancing or periodic vibration checks. *These transducers are limited in frequency response to about 12,000 CPM (200 Hz) and should not be used for shaft surface speeds over approximately 15,000 ft/min (approximately 5000 m/min).*

2.3241 Shaft Sticks

A shaft stick should only be used if a non-contacting displacement probe has not been installed in a large journal bearing machine. Figure 37 is a picture of a shaft stick in use. It consists of a hardwood, "fish-tailed" stick outfitted with a stud on its other end to attach a velocity pickup or an accelerometer. The fish-tail shape helps keep the shaft stick in contact with the shaft and the edge of the fish tail is tapered to reduce friction and prevent chatter. The shaft stick is hand-held perpendicular to the shaft.

When using the shaft stick, use special care to follow these precautions:

- Do not mount the stick on shafts traveling faster than approximately 15,000 ft/min (5000 m/min) for safety reasons.
- Try to use the same pressure from one survey to the next when taking PMP type measurements as this can affect your month to month vibration amplitudes.
- Do not rely on the shaft stick to monitor "high frequency" defects such as bearings, blade passes, gear mesh frequencies, and certain electrical problems.

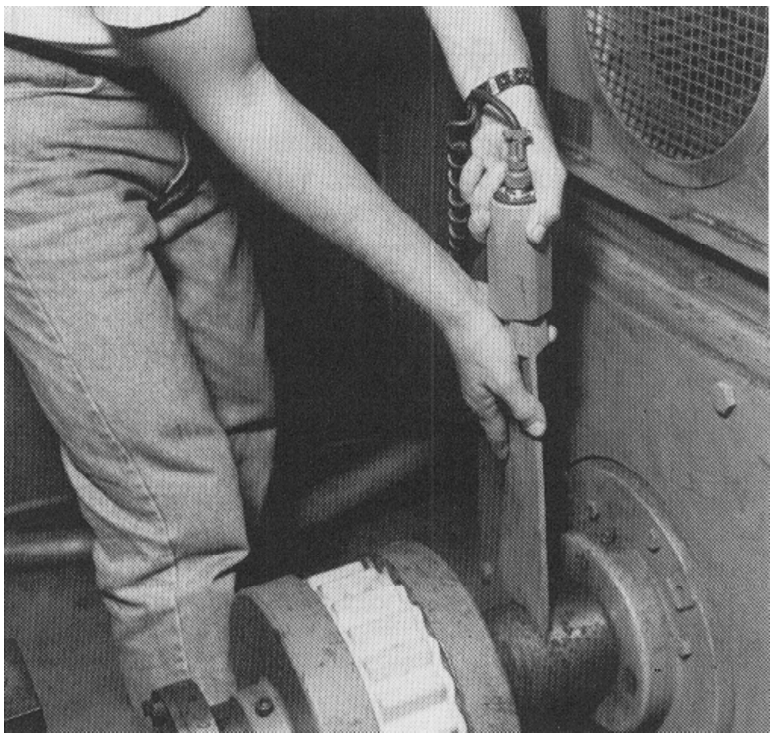

FIGURE 37
SHAFT STICK (Ref. 1)

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2.3242 Shaft Riders

A shaft rider (pictured in Figure 38) is mounted permanently in the housing and extends into the casing to measure shaft displacement by using a spring mounted tip that rides on the shaft's surface. One of two methods is usually used. In the first, an accelerometer or velocity transducer is mounted on the end of the shaft rider (quite similar to using an accelerometer or velocity transducer with a probe). In the second, a spring-loaded rod is driven by the vibration and moves a mass inside coiled wires surrounded by magnets. An electrical signal which is used to measure the vibration is thereby generated.

Once again, the shaft rider is an older technology which has gradually been replaced by the non-contact eddy current probe, but is still in place on many of today's turbine/ generators. Typically, the shaft rider is employed to continually monitor the vibration of the shaft in order to determine if the vibration is at a level which would signal that a machine should be shut down or that an alarm should be sounded. Like the shaft stick, its usable frequency range is up to about 12,000 CPM. Furthermore, *it should be used only on machines that do not have shaft speeds which exceed 3600 RPM and a shaft surface speed below approximately 20,000 ft/min. Thus, the shaft rider is useful only for monitoring unbalance, misalignment, and other problems occurring at 1X thru 3X RPM. It will not be able to detect rolling element bearing wear or failures, gear mesh problems, and other high frequency problems.*

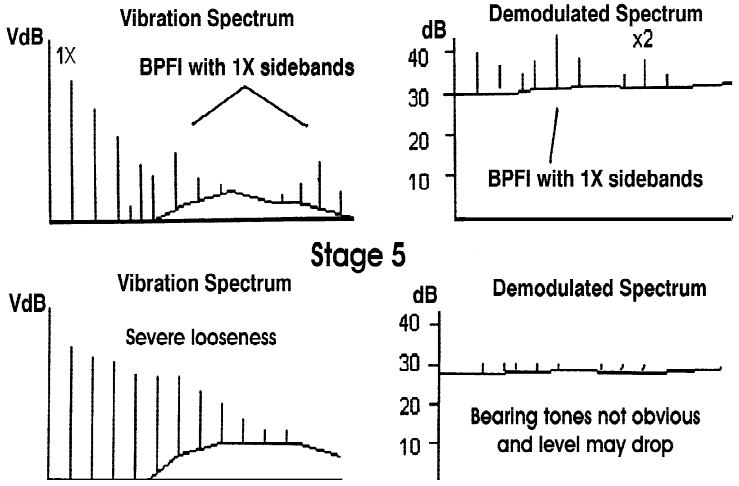

FIGURE 38
SHAFT RIDER (Ref. 5)

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2.33 SELECTION CRITERIA FOR TRANSDUCERS

Below are the descriptions of the items in Table III, plus other items which should be considered when selecting a transducer:

1) SENSITIVITY RANGE - Sensitivity is the capability of the transducer to determine the amplitude of vibration (displacement, velocity, or acceleration) from the amplitude of the voltage signal. For example, an accelerometer may have a sensitivity of 100 mV/g. This means that if this 100 mV/g accelerometer saw 10 mV at a frequency, it should convert this voltage to an amplitude of 0.1 g (10/100 = 0.1).

2) FREQUENCY RANGE - Frequency range is a measuring capability of the transducer from a low limit to a high limit of frequency. Each transducer has its own frequency range (which must be known by the user). Typically, the transducer's frequency response is given at various amplitude tolerances, such as ±5%, ±10%, and/or ±3dB.

3) NATURAL FREQUENCY RANGE - The natural frequency of the transducer is the factor that limits the upper end of the frequency range of the transducer. Therefore, it is very important to choose an accelerometer in which the highest frequency of interest is well below its natural frequency. Normally, the effective accurate range of its frequency will be about 20% - 33% of its stud-mounted natural frequency. Thus, for an accelerometer with a natural frequency of 30,000 Hz (1,800,000 CPM), the effective range for the upper frequency will be limited to about 6000 - 10,000 Hz (360,000 - 600,000 CPM).

4) TRANSDUCER MASS - The weight of the transducer must be only a fraction of the weight (usually below 10%) of the housing to which it will be attached. If the transducer is heavy enough to affect the mass, it can change the natural frequency of the system, and therefore, the true frequency response.

5) USABLE TEMPERATURE RANGE - This is the minimum and maximum temperature that a transducer can withstand without significantly affecting its response capabilities. This is especially important when selecting transducers to be mounted permanently on machinery that is subjected to very high or very low temperatures. Also, please see the important comments in Section 2.321 pertaining to how differently shear mode accelerometers respond to temperature *differences* than do compression mode accelerometers, particularly when taking low frequency measurements (see these accelerometer types in Figures 17 thru 19).

6) MEASUREMENT DIRECTION - Most transducers measure only in the mounting direction (with only a small percentage reaction to vibration in directions perpendicular to the mounting - typically 3% to 6%). However, triaxial transducers measure vibration in all three directions simultaneously.

7) TRANSDUCER SIZE - The measurement location on a machine may require a transducer to be of a certain size or cross section to physically fit on the measuring surface. For instance, on a small surface, a small transducer must be used. However, due to its small size, the transducer may have a low voltage sensitivity restricting its capability to pick up low frequency data.

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8) TRANSDUCER POWER SUPPLY - There are 3 common methods to power a transducer:

i) ICP[15] (Integrated Circuit Piezoelectric) Power Supply. The vibration instrument sends the power through the cable to the transducer.

ii) Independent Power Supplies. A battery (DC) or AC-powered unit is used to send power to the transducer.

iii) Charge Amplifier. A battery or AC powered unit is placed between the transducer and the vibration instrument to amplify the signal voltage.

9) CABLING - ICP powered transducers can generally use cables up to about 1000 feet in length. For transducers which are not ICP powered, a charge amplifier may be required at approximately 50 foot intervals of cable length (or less). Discuss the cabling requirements for the transducer with the transducer vendor.

10) MOUNTING SENSITIVITY - As described in **Sections 2.34 and 2.341**, there are many ways to mount transducers (hand-held probes, magnetic connectors, permanent stud mounts, adhesive mounts, etc.). Each has a significant effect on the ability of the transducer to measure the vibration accurately, as well as to reproduce the same spectra in subsequent measurements. ***This one fact is often critical to obtaining accurate, repeatable data (and, therefore, to the success or failure of the entire PMP program)***.

11) MAGNETIC INTERFERENCE - Except for piezoelectric accelerometers, magnetic interference affects the performance of transducers. Since eddy current probes rely on a magnetic field to determine shaft gap, the influence from another magnetic field will produce erroneous readings. Likewise, moving coil seismic velocity pickups can be effected by magnetic fields. Even cable movement generates a magnetic field that can distort the signal in the cable. The magnetic field around large generators and motors can affect seismic transducers, cables, and even the analyzer itself.

12) SEALING METHOD - I*f they are not certified as being hermetically sealed, all accelerometers will "micro-breathe", allowing moisture to affect their lower frequency (below 3000 CPM or 50 Hz) performance dramatically*. If the sealing method is not listed in the specifications, contact the accelerometer vendor and verify.

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2.34 MOUNTING OF TRANSDUCERS (ACCELEROMETERS)

There are 5 typical transducer mounting methods, as listed in Table IV. Each mounting method will provide a typical frequency response range when transducers capable of operating within these frequency response ranges are used.

TABLE IV
TRANSDUCER MOUNT USABLE FREQUENCY RANGE FOR THE WILCOXON 726T ACCELEROMETER (Ref. 9)

ACCELEROMETER MOUNTING	MAXIMUM ACCEPTABLE FREQUENCY (CPM)	MOUNTING NATURAL FREQUENCY (CPM)
1) Stud Mount	975,000	1,900,000
2) Adhesive Mount with Hottinger Baldwin Messtechnik X60	540,000	None Observed
3) Stud Mount on Rare Earth Magnet	450,000	724,500
4) Mounted on Quick-Connect Stud Mount	360,000	609,000
5) Hand-held Mount Using a 2" Probe	48,000	88,500

Table IV is a summary of an article written by Computational Systems, Inc. (CSI)[9]. Many other factors come into play when the effectiveness of a transducer mounting type is considered. Refer to the referenced article for an in-depth look at transducer mounting effectiveness. Figure 39 shows illustrations of these mounting methods (Quick-Lock not shown).

2.341 Transducer Mounting Applications

Each particular transducer mounting method has certain applications which will be discussed in detail. Following below is a description of each mount, along with some important comments which should be made *(remember that an incorrect transducer mount, such as use of a hand-held probe can corrupt the data you are trying to acquire, and can even miss very important high frequency data pertaining to gears, rolling element bearings and electrical problems)*:

STUD MOUNT - Stud mounting is used for permanently mounted transducer applications. Sometimes, an adhesive will be used in combination with stud mounting to prevent the transducer from working its way off of the stud mount. While stud mounting is not practical for collecting PMP route data, *if very high frequency measurements (>3,000,000 CPM or 50,000 Hz) are required,* it will have to be used at those data collection points. Also, *stud mounting gives extremely repeatable data from measurement to measurement from one survey to the next and provides the highest frequency response range.*

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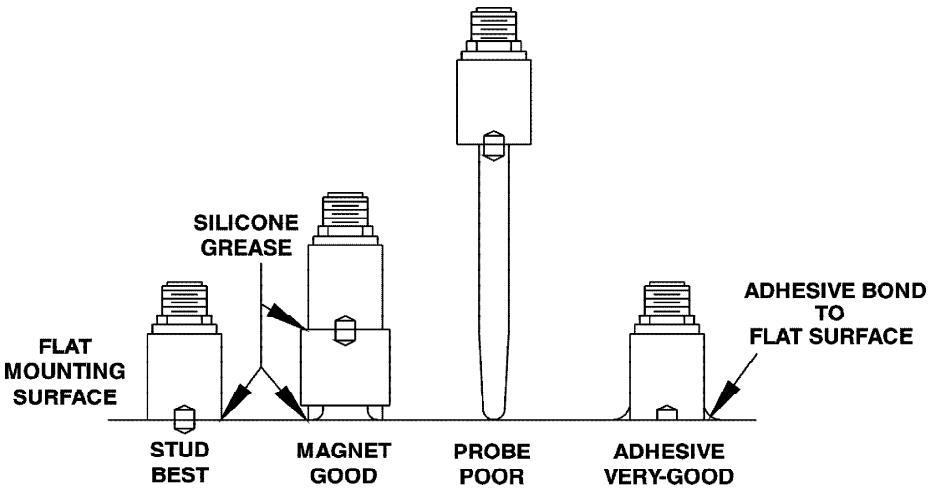

FIGURE 39
MOUNTING METHODS ILLUSTRATED (Ref. 1)

ADHESIVE MOUNT - Adhesive mounts are very useful in diagnostics where reliable high frequency data is required since a stud mount is rarely found at an exact point where a measurement is needed. *Occasionally, adhesive mounts are used on a PMP route to collect high frequency data (>2,400,000 CPM or 40,000 Hz)*. Therefore, great care should be taken when using an adhesive to permanently mount a transducer because these mounts may work loose over long periods of time. *Adhesive mounts provide a very good frequency response range if the proper adhesive type is used and if only a thin layer of adhesive is applied.* Also, *adhesive mounting gives very repeatable data over a series of measurement surveys.*

MAGNETIC MOUNT - *Magnetic mounts are the most common used method in PMP programs today as well as when taking diagnostic data. The upper frequency response range is generally from about 120,000 to 450,000 CPM (depending on the magnet type, its pulling force, and especially how well the magnet mounts on the surface).* Because some machines such as centrifugal compressors have gear mesh frequencies and harmonics that range from 900,000 to 4,000,000 CPM (15,000 - 65,000 Hz), data on these machines must be taken with an adhesive or stud mount instead. *However, a magnetic mount tends to give reliable repeatability over measurement surveys which is adequate for PMP purposes when used within the frequency ranges applying to that particular magnet (it is most important to verify this with the vendor supplying the transducer and magnet).* Also, frequency response of a magnet mounted accelerometer can be greatly enhanced at frequencies exceeding approximately 90,000 CPM (1500 Hz) if one takes 10 to 15 seconds to insert silicone grease between the magnet and the accelerometer. This was clearly illustrated in the *Sound and Vibration* magazine article authored by Dr. Ken Piety *et al* listed in Reference 9.

QUICK-CONNECT MOUNT - *Quick-Connect mounts are also ideal for collecting PMP route data* since they provide easy mounting and dismounting. They have a relatively large frequency range which is able to detect most common machinery problems which occur in higher frequencies such as bearing defects. However, a quick-connect mount *should not be used for detecting high frequencies above approximately 420,000 CPM* such as those found in high speed compressor gear mesh frequencies and harmonics (see Ref. 9). *Nevertheless, repeatability between measurement surveys is consistent enough for PMP purposes for frequencies below this limit.*

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HAND-HELD PROBE - *This is the most undesirable mounting method.* The *usable frequency range is only up to a maximum of 60,000 CPM, regardless of the length, diameter or material of the probe! (as per Ref. 9). Depending on the length of probe used, the maximum accurate frequency measurable may only be 30,000 CPM.* With these constraints, *the only machinery problems detectable and repeatable will normally be those which occur at 1X, 2X, and 3X RPM for machinery running under 3600 RPM or less. Most importantly, bearing defect frequencies and their harmonics, and bearing component natural frequencies often lie above 30,000 CPM. They can be missed entirely if a hand-held probe is used.* Furthermore, as demonstrated in the research covered in Reference 9, the *repeatability of measurement amplitudes* and maximum frequency ranges *from survey to survey is not consistent* and will vary depending on the probe position or amount of hand pressure applied as well as on which person is taking the data. This impairs the analyst's ability to accurately trend problems and *may cause him to miss some problems that occur at the top (or beyond) of the probe's measurable frequency range.* Hand holding a transducer or probe is useful in hard-to-reach places such as where a screen prevents the use of a magnet or other mount type. Also, it is useful if safety is a concern and the analyst cannot safely reach into moving parts of the machine. However, be aware that *using this method will miss very important information that is crucial to maintaining a successful PMP program.* In the instances described above, it is wiser to attach a permanently mounted transducer in unsafe or hard to reach locations to obtain useful diagnostic data. *Furthermore, this example shows that probes actually can add "phantom" data in the bearing defect range which makes the bearing appear to be in its last stage of failure due to the natural frequency of the probe being excited even though there may be no bearing wear at all.*
And, what might be worse is that Reference 9 proved even the short 1/2 inch probes will almost always totally miss data at frequencies exceeding 120,000 CPM (2000 Hz) which again is where many significant gear, rolling element bearing and electrical problems are so often detected (sometimes with little or no evidence of such problems below this frequency)*.*

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2.4 UNDERSTANDING VIBRATION PHASE AND ITS APPLICATIONS

2.41 INTRODUCTION

Phase is the timing relationship vibration has with respect to another vibrating part or fixed reference point. It can also be thought of as the vibration motion at one location relative to the vibrating motion at another location (for example, phase at the outboard bearing in the horizontal direction relative to that in the vertical).

Phase is easiest to visualize if one is familiar with using a timing light to set the timing of an automobile engine. Vibration phase is very similar except that the vibration (not the spark) is the trigger. *If an analyst clearly understands phase, he can use this powerful analytical tool to differentiate between the many problems which can cause high vibration at 1X RPM, 2X RPM, and 3X RPM with conviction*. However, the vibration frequencies have to be the same or exact multiples to properly compare phase differences. By taking phase measurements on each bearing housing in each direction (horizontal, vertical and axial), he can determine whether the problem is unbalance, misalignment, soft foot, bent shaft, eccentric rotor, loose hold-down bolts, resonance, cocked bearing or several other potential problems (all of which can generate vibration spectra which appear to be *identical*). In a later chapter section, it will be shown that a trigger will be required to serve as a reference when using a single-channel analyzer to acquire phase measurements.

When should phase measurements be taken? This section, along with the chapter describing each of the items in the **"Illustrated Vibration Diagnostic Chart"** will show that *phase measurements should be taken when high vibration is found at 1X RPM, 2X RPM and/or 3X RPM*. In other words, phase measurements are not intended to be taken at each point on each machine during routine data collection.

2.42 HOW TO MAKE PHASE MEASUREMENTS

Figures 40 and 41 illustrate two typical strobe light methods for taking phase measurements (strobe light for use with data collectors are now available). Another method, shown in Figure 42, utilizes a stationary photocell or laser-tach targeted at a piece of reflective tape mounted to the rotating part.

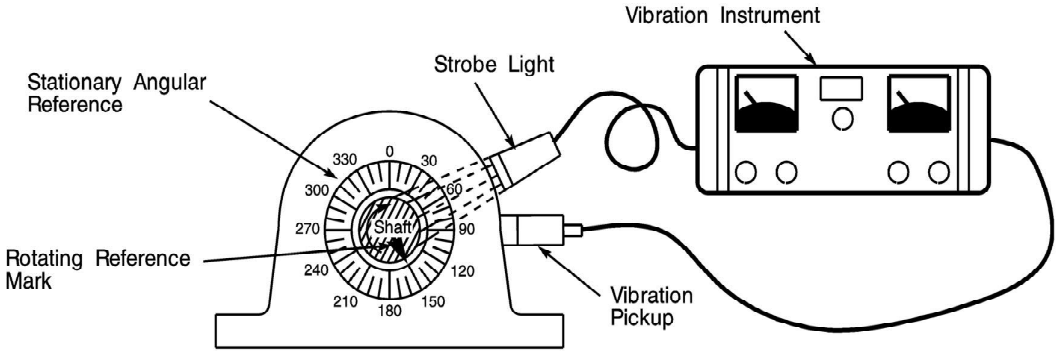

FIGURE 40
PHASE REFERENCE WITH A ROTATING REFERENCE MARK AND A STATIONARY ANGULAR REFERENCE

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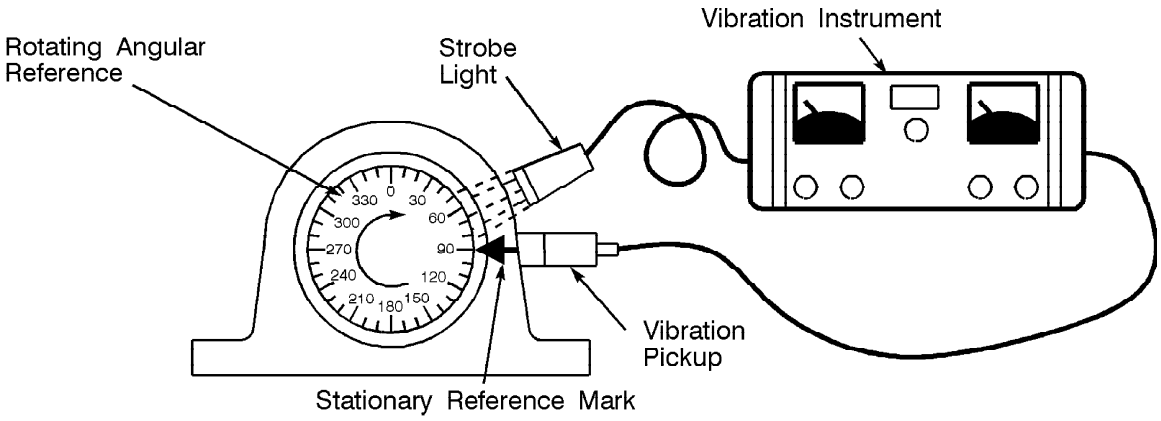

FIGURE 41
PHASE MEASUREMENTS WITH A STATIONARY REFERENCE MARK AND ROTATING ANGULAR REFERENCE

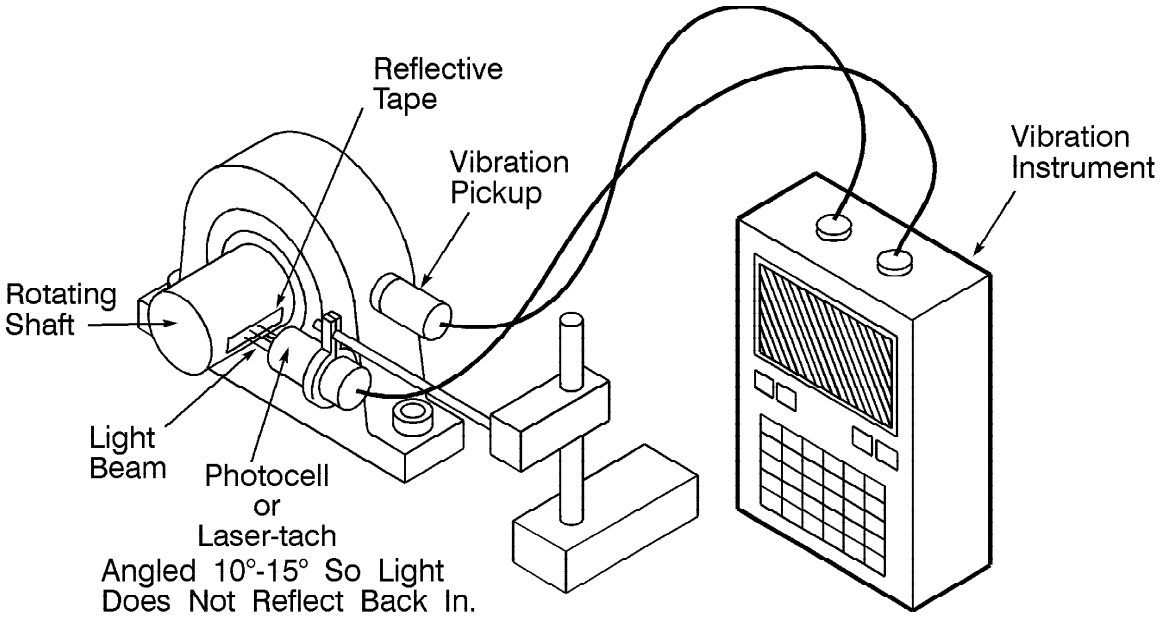

FIGURE 42
PHOTOCELL METHOD OF ACQUIRING PHASE MEASUREMENTS

To properly collect phase measurements, it is important that the following be done first. After the strobe light is "tuned" to the frequency of interest, the vibration amplitude and phase must be recorded. If using a swept-filter instrument like that shown in Figure 40, the strobe light can be oriented in any way to best see the reference mark, but the transducer must be firmly mounted in place at each measurement point. *In contrast, if a photocell is used, both the photocell and the transducer must be locked down at each measurement point*. Only the transducer gets moved to the next location.

The photocell method is more accurate than the swept-filter strobe method since the instrument measures the phase angle within very accurate tolerances. Since the strobe method is hand-held, it includes human error.

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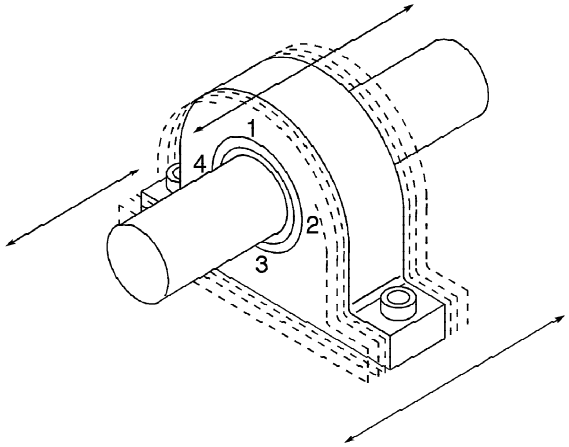
2.43 USING PHASE ANALYSIS IN VIBRATION DIAGNOSTICS

To determine whether the whole bearing face is moving back and forth axially (as in Figure 43), or if it is vibrating with a twisting motion (as shown in Figure 44), phase measurements are taken from four points on the bearing housing in the axial direction.

2.431 Evaluating Axial Motion of a Bearing Housing to Reveal a Possible Cocked Bearing or a Bent Shaft

Using the conventions shown in Figure 43, the transducer should be mounted at locations 1, 2, 3, and 4. In this case, the phase analysis indicates that the bearing is moving axially in a planar motion.



PHASE	
1	150°
2	150°
3	150°
4	150°

FIGURE 43
AXIAL PHASE SHOWING PLANAR MOTION

However, if the same four measurements produce a 90° phase difference at each location, as shown in Figure 44, a bent shaft or a cocked bearing would be suspected (particularly if confirmed with axial phase measurements on the other bearing of the same rotor as in Figure 45. In this case, the 180° phase change across points 1 and 3 indicates an up-and-down twisting motion while the 180° difference between points 2 and 4 reveals a side-to-side twist.

PHASE	
1	60°
2	150°
3	240°
4	330°

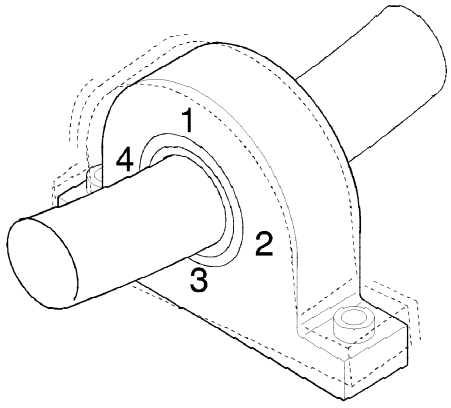


FIGURE 44
AXIAL PHASE SHOWING TWISTING MOTION DUE TO BENT SHAFT OR COCKED BEARING

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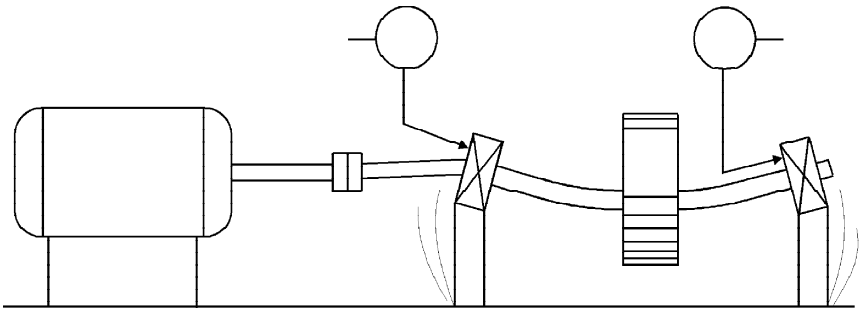

FIGURE 45
DETECTION OF BENT SHAFT PROBLEMS

Bent shaft problems cause high axial vibration with axial phase differences tending toward 180° on the same machine component (that is, between outboard and inboard bearings of a motor or between the driven rotor bearings themselves). Dominant vibration normally occurs at 1X if bent near shaft center, but at 2X if bent near the coupling. (Be careful to account for transducer orientation for each axial measurement if you reverse transducer direction.)

2.432 Phase Behavior Due to Unbalance

To check for an unbalance condition, use the radial phase measurements, as illustrated in Figure 46. I*f an unbalance condition is the only problem a machine has, the phase increments between each of the radial measurement locations on each bearing will be 90° apart.* If the transducer is moved 90° at a time, significant departure from 90° phase difference means that the problem is something other than unbalance. *An even better indicator of unbalance is to compare the difference in phase in the horizontal direction on the outboard and inboard bearings. Then, find the vertical phase difference between measurements on the outboard and inboard bearings. If the unbalance is significant, the horizontal phase difference should equal the vertical phase difference between the outboard and inboard bearings within ± 30°. Also, the 1X RPM peak will be high*. This shows that the resultant motion of the rotor is the same in both the horizontal and vertical directions. If the resultant motion is not the same, the *dominant problem* is something other than unbalance.

2.433 Phase Behavior Due to Looseness/Weakness

If the amplitude and/or phase changes appreciably between the mated components, as shown in Figure 47, looseness/weakness should be suspected. Note the significant amplitude and phase change between the measurements on the baseplate and the supporting concrete base. This problem may be caused by inadequate grouting between these two surfaces. *When phase changes across any two interfaces* (i.e.., between the base plate and concrete base) *exceed approximately 90°, such a problem is usually indicated.* Also, look for great differences in amplitudes between each interface (although this can occur if there is considerably more mass and, therefore, less vibration response at one of the interfaces).

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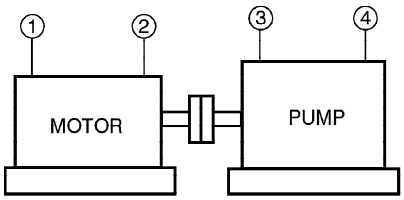

TABLE A.
PHASE AT 1X RPM FOR A MACHINE
WITH DOMINANT FORCE UNBALANCE

DIR.	1	2	3	4
A	60°	*70°	60°	*80°
H	30°	25°	30°	40°
V	120°	110°	120°	135°

* Corrected Phase Values Accounting For 180° Orientation of Accelerometer.

TABLE B.
PHASE AT 1X RPM FOR A MACHINE
WITH DOMINANT COUPLE UNBALANCE

DIR.	1	2	3	4
A	60°	*70°	60°	*80°
H	30°	210°	200°	220°
V	120°	295°	280°	300°

* Corrected Phase Values Accounting For 180° Orientation of Accelerometer.

TABLE C.
PHASE AT 1X RPM FOR A MACHINE
WITH SIGNIFICANT DYNAMIC UNBALANCE

DIR.	1	2	3	4
A	60°	*70°	60°	*80°
H	30°	90°	80°	70°
V	120°	180°	170°	165°

* Corrected Phase Values Accounting For 180° Orientation of Accelerometer.

FIGURE 46
TYPICAL PHASE MEASUREMENTS WHICH WOULD INDICATE EITHER STATIC, COUPLE OR DYNAMIC UNBALANCE

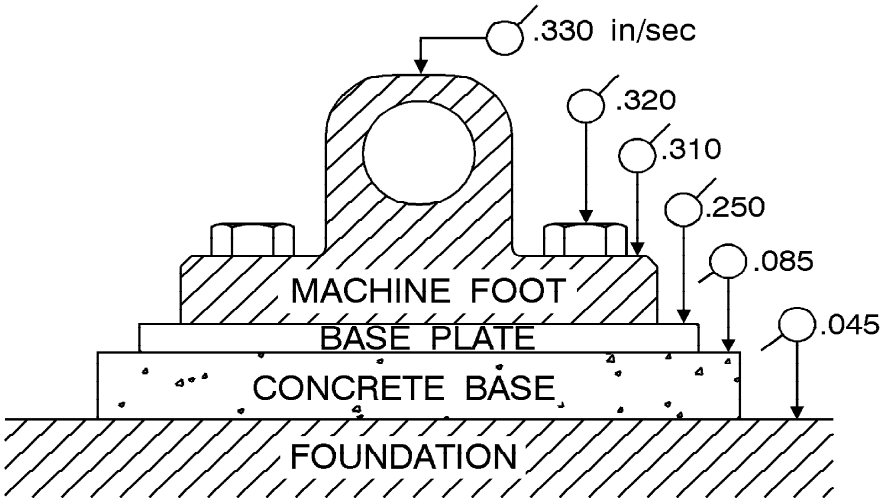

FIGURE 47
PHASE MEASUREMENTS FOR LOOSENESS

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2.434 Phase Behavior Due to Misalignment

Figure 48 illustrates the two basic types of shaft misalignment.

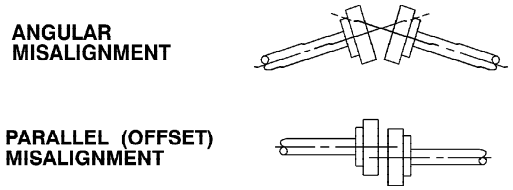
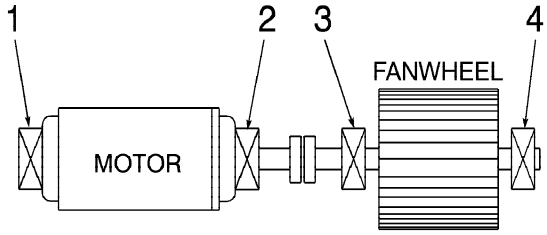


FIGURE 48
DIAGRAM OF ANGULAR AND PARALLEL (OFFSET) SHAFT MISALIGNMENT

Most misalignment is rarely pure angular, or pure parallel, but usually a combination of both instead. Shaft misalignment is characterized by phase differences approaching 180° across the coupling. Phase differences are *not exactly* 180° out-of-phase or 0° in-phase, but usually within 30° of each other. The closer the phase difference is to 180°, the higher is the probability of misalignment. Radial phase readings are normally more sensitive to parallel misalignment and axial phase readings to angular misalignment.

Amplitude and phase measurements should be taken both axially and radially on both bearing housings on both sides of the coupling. Then, the phase readings should be compared - first, between the bearing housings on each side, then between the housings on each side of the coupling itself. A phase difference of 180° across the coupling strongly indicates shaft misalignment.

To check for angular misalignment in particular, readings should be taken in the 4 quadrant locations like the conventions shown in Figures 43 and 44 (or in as many of the 4 points as accessible). An example of this is given in Figure 49. Consider the data which is provided. Motor Bearings 1 and 2 are moving in unison with one another while Bearings 3 and 4 are also moving together (this indicates there is no internal misalignment, bent shaft, or cocked bearings in either the motor nor the fan). *However, notice the 180° phase change across the coupling*. *This indicates that the motor and fan shafts are misaligned between Bearings 2 and 3*, and show that, at any instant of time, the motor shaft is always moving in a direction opposite to that of the fan shaft in the axial direction.



1	2	3	4
1 90°	1 90°	1 270°	1 270°
2 105°	2 105°	2 270°	2 255°
3 105°	3 90°	3 270°	3 270°
4 90°	4 85°	4 270°	4 270°

Note: Phase measurements have been corrected for transducer direction.

FIGURE 49
AXIAL PHASE COMPARISONS FOR ANGULAR MISALIGNMENT

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Since the transducer often must be turned 180° when making some of the axial measurements, *remember to add or subtract 180° from the phase reading each time it is oriented opposite the reference direction* (or viewing) that was chosen, so that the phase readings are forced to orient in the same reference direction.

2.435 Using Phase Analysis to Find the Operating Deflection Shape of a Machine and Its Base

Acquisition of the operating deflection shape can go far in helping an analyst diagnose both the cause and severity of problems on a machine or support structure. As the name implies, this technique *reveals how the member is vibrating at a certain frequency.* Importantly, *operating deflection shape analysis is not equivalent to a modal analysis* since it does not require acquisition of the frequency response functions (FRF's) of a machine member. And, it can be performed using only a once per revolution trigger and a single channel instrument (even a swept-filter analyzer and strobe light). *While it often may reveal resonant operation, it just as often can detect misalignment, soft feet, etc*.

The operating deflection of a machine and/or its support structure can be determined by dividing the structure into 10 or 12 equally-spaced measurement locations and recording the phase and vibration values at each location. The objective is to determine the location of the maximum and minimum flexure. With this information, coupled with the forcing frequency (1X RPM, 2X RPM, or at 6X RPM in the case of a 6-vaned pump impeller), a solution can be derived to reduce the excessive motion if resonance is found by adding braces or mass, by changing the forcing frequency (or by replacing anchor bolts or correcting "sprung feet" if excessive vibration is found in a support foot or two).

Figure 50 shows some equally spaced phase measurement locations on a machine support frame which had an apparent flexing problem at 1X RPM in the vertical direction of the machine mounted on it. *If only the vibration amplitude was used* without the phase information, a plot of the vibration amplitude on a drawing of the frame structure may appear like the one shown in Figure 51.

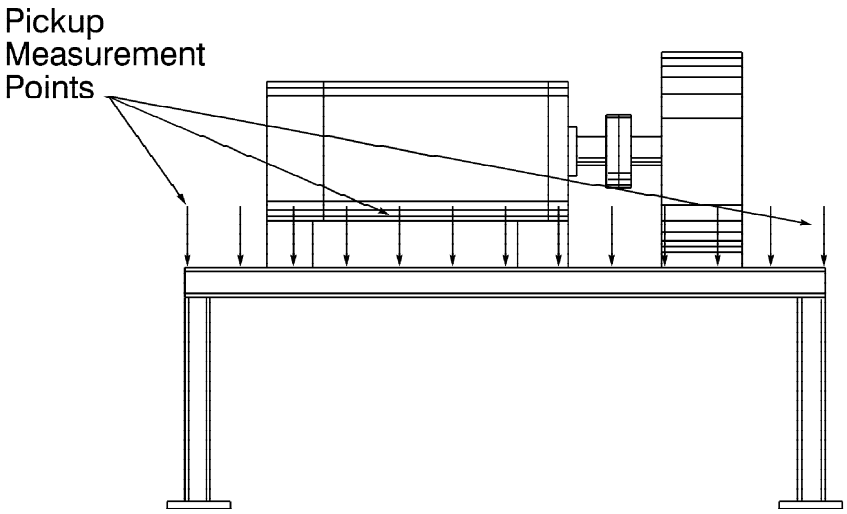

FIGURE 50
PHASE MEASUREMENT LOCATIONS ON FLEXING MACHINE BASE USED TO DETERMINE ITS OPERATING DEFLECTION SHAPE

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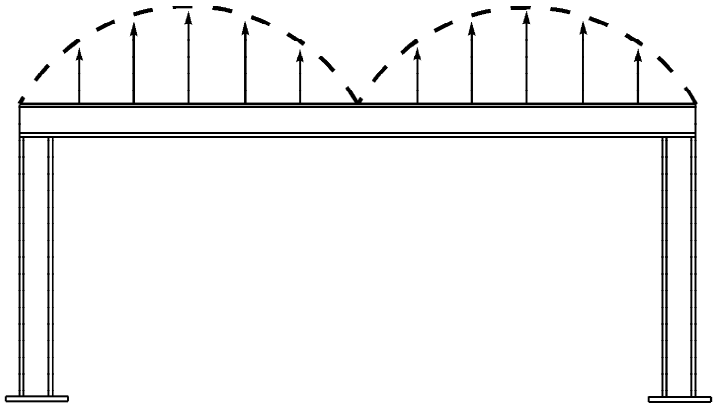

FIGURE 51
PLOTTING ONLY AMPLITUDE WITHOUT PHASE WILL CREATE AN INCORRECT OPERATING DEFLECTION SHAPE

However, by including the phase information, the true operating deflection shape is shown in Figure 52. The phase data verifies that the left and right halves of the structure were actually moving in opposite directions. Also, note the significant phase change of 165° (270° - 105°) measured just to either side of mid-span. In this particular case, this was a strong indication the support frame was being excited at its second bending mode. This was subsequently verified by a series of natural frequency tests.

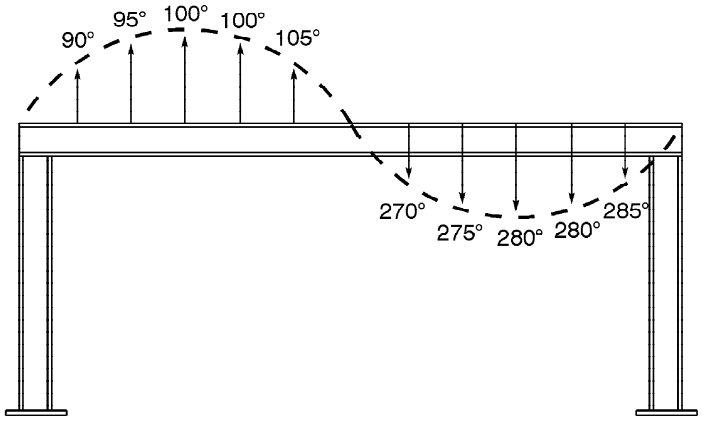

FIGURE 52
ADDITION OF PHASE CREATES AN ACCURATE OPERATING DEFLECTION SHAPE DRAWING

This type of structural analysis is very important because it shows that stiffening the center of the span, as shown in Figure 53, would have very little effect since it is at a node where the vibration is minimum.

A more effective location for such bracing would be at an antinodal point where the vibration was the greatest, as shown in Figure 54. This bracing was added at the location shown in Figure 54. As a result, the vertical second bending mode increased in frequency by 16%. As a result, vibration levels dropped dramatically from approximately .82 in/sec to only .12 in/sec peak (-85%).

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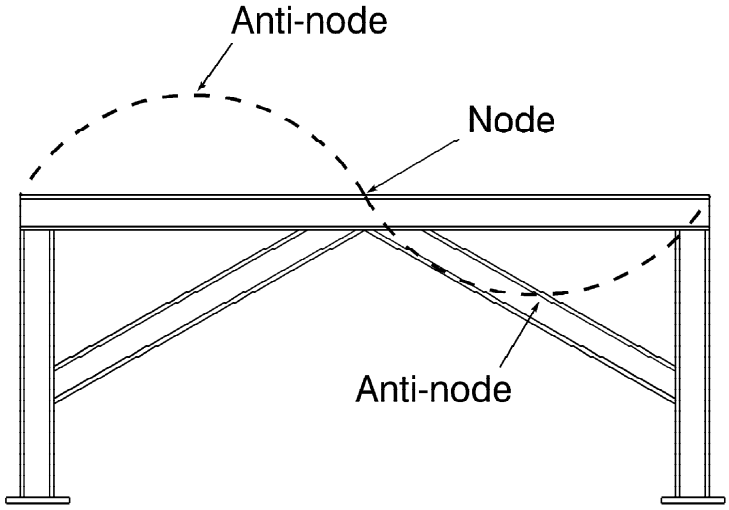

FIGURE 53
INEFFECTIVE APPLICATION OF STIFFENING AT A NODE

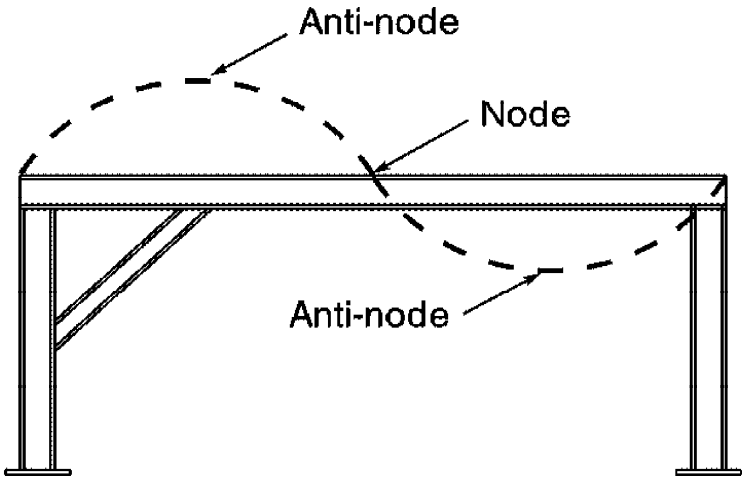

FIGURE 54
STIFFENING AT THE ANTI-NODE AS DETERMINED BY THE MODE SHAPE

Operating deflection shape software is now available which will automate the measurement process and allow a computer to animate the operating shape on a monitor. This makes this technique much more powerful, particularly when attempting to show the results to a person who is not experienced in vibration analysis.

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8. Spectral Dynamics (a Scientific-Atlanta Division); San Diego, CA; Vibration Handbook; 1990.

9. Bowers, S. V.; Piety, K. R.; and Piety, R. W.; Computational Systems, Inc.; Knoxville, TN; "Real-World Mounting of Accelerometers for Machinery Monitoring"; Sound and Vibration Magazine; February, 1991; pages 14-23.

10. Computational Systems, Inc. (CSI); Knoxville, TN; "Using the CSI 2110 With Supervisory Systems"; TrendSetter; May, 1991; Vol. 2, No. 3; page 3.

11. "Spike Energy Spectrum" - developed by IRD Mechanalysis, Inc., based in Columbus, OH.

12. "Amplitude Demodulated Spectrum" - developed by Computational Systems, Inc. (CSI), based in Knoxville, TN.

13. "Acceleration Enveloped Spectrum" - developed by SKF Condition Monitoring with headquarters in San Diego, CA.

14. "SEE Spectrum" - developed by SKF Condition Monitoring with headquarters in San Diego, CA.

15. "ICP" is a trademark established by PCB Piezoelectronics, headquartered in Depew, NY. "ICP" stands for "Integrated Circuit Piezoelectric".

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APPENDIX
SPECIFICATIONS FOR VARIOUS TRANSDUCERS FROM A VARIETY OF MANUFACTURERS

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Model 970
Vibration Pickup

SPEC SHEET: 41-06-04
EFFECTIVE: May 1992

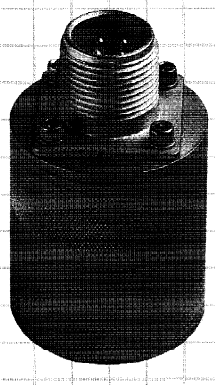
DESCRIPTION

The IRD Model 970 is a rugged low impedance accelerometer type vibration pickup for use with portable instruments on all types of rotating machinery.

The Model 970 has built-in electronics to provide low impedance output and eliminate the need for special low noise cables. The Model 970 is sealed against dust and moisture and operates up to a temperature of 250° F (121°C).

A single 1/4-28 mounting stud is used for attaching the pickup to a flat machine surface. Insulated Mounting Adapters are available for those cases where ground potential problems may exist between the pickup mounting surface and instrument.

The Model 970 connects directly to IRD Model 810, 840, 880 and other IRD instruments designed to accept the 970. Operating power is obtained from the instrument.



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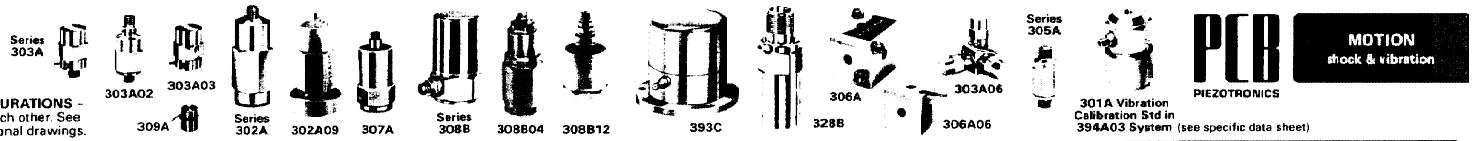
ICP QUARTZ
ACCELEROMETERS
with built-in amplifiers[1]
Series 300

STANDARD DESIGN CONFIGURATIONS - Photos are not proportional to each other. See page following chart for dimensional drawings.



PCB PIEZOTRONICS — MOTION shock & vibration

	MINIATURE, LOW MASS					GENERAL PURPOSE - HIGH SENSITIVITY															INDUSTRIAL		TRIAXIAL			SHOCK				
Application	Stud Mtg Side Conn	Stud Mtg Top Conn	Adh Mtg Side Conn	High Sens 100mV/g	One★ Gram Weight	Low Strain	Gnd★ Iso, Lo Strain	Gnd★ Iso 10K	Integral★ Mtg Base B Strain	Hi Sens Lo Mass Gnd Iso	High Sens	High★ Freq Low Mass	Side Connector		Top Conn Gnd Iso	High Temp ±325°F	2nd Order LP Filter★ Built-in	Seismic		Low Strain	Herm★ Seal Gnd Iso	One Inch Cube	5/8★ Inch Cube	Three 305A's on Block	General Purpose	High g, Low Mass			
																		Low Mass	Low Freq										
Model Number (see optional models below specifications)	**303A**	**303A02**	**303A03**	**303A11**	**309A**	**302A**	**302A04**	**302A06**	**302A09**	**308B15**	**302B03**	**307A**	**308B**	**308B03**	**308B12**	**308B08**	**308B30**	**368B02**	**393C**	**308B04**	**328B**	**306A**	**306A06**	**303A06**	**302A02**	**305A**	**305A02**	**305A03**	**305A04**
Sensitivity Peak[3] mV/g	10	10	10	100★	5	10±2%	10±2%	10±2%	10±2%	100±2%★	300±2%★	100±5%★	100±2%	50±2%	100±5%	100±2%	100±2%	1000±5%	1000	100±2%	100±5%	10±2%★	10	10	10±2%	.05	.1	.5	1
Range (for ±5V out) ±gpk	500	500	500	10 (±1V)	1000	500	500	500	100	50	16	50	50	100	50	50	50	5	2.5 ±2.5V	50	50	500	500	500	500	100 000★	50 000	10 000	5000
Range (for ±10V out)[11] ±gpk	—	—	—	—	—	1000	1000	1000	—	—	32	100	100	200	100	100	100	10	—	100	100	1000	—	—	1000	—	—	20 000	10 000
Resolution gpk	.01	.01	.01	.005★	.02	.01	.01	.01	.005	.001	.0025	.005	.001	.002	.001	.001	.001	.0005	.0001★	.001	.001	.01	.02	.01	.01	2	1	2	1
Resonant Freq (mtd) >kHz	70	100★	70	70	120★	45	45	45	20	25	20	>40	25	25	30	25	25	25	3.5	25	18	8	28	7	45	60	60	60	60
Freq Range ±5%[7] Hz	1 to 10 000	1 to 10 000	1 to 10 000	5 to 10 000	5 to 10 000	1 to 5000	1 to 5000	1 to ★ 10 000	1 to 5000	1 to 3000	1 to 5000	2 to 10 000	1 to 3000	1 to 3000	1 to 3000	4 to 3000	1 to 20	2.5 to 3000	.025 to 800★	2 to 5000	1 to 3000	1 to 1000	1 to 3000	1 to 1000	.05 to 5000	.25 to 8000	.25 to 8000	.25 to 8000	.25 to 8000
Freq Range ±10%[2] Hz	7 to 20 000	7 to 20 000	7 to 20 000	3 to 20 000	2 to 20 000	7 to 10 000	7 to 10 000	—	7 to 10 000	7 to 6000	7 to 7000	1.3 to 15 000	7 to 6000	7 to 6000	7 to 5000	2 to 6000	7 to 24	1.5 to 5000	.01 to 1200	1.3 to 7000	7 to 5000	7 to 3000	7 to ★ 10 000	—	.03 to★ 10 000	.1 to 12 000	.1 to 12 000	.1 to 12 000	.1 to 12 000
Overload Recovery μs	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10
Discharge Time Constant[2] @ 70°F >s	0.5	0.5	0.5	0.2	0.1	0.5	0.5	0.5	0.5	0.5	0.2	0.3	0.5	0.5	0.5	0.2	0.5	0.2	20	0.5	0.5	0.5	0.5	0.5	10	2.0	2.0	2.0	2.0
Amplitude Linearity (zero based best straight line) %	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
Output Impedance★ <ohm	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100
Transverse Sens (max) %	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5
Strain Sens g/μin/in	.05	.05	.05	.05	.005	.01	.01	.01	0★	.05	.01	.1	.05	.05	.05	.05	.05	.05	.001	.01	.1	.01	.05	.01	.01	.2	.2	.2	.2
Temperature Range[8] °F	-40 to +200	-40 to +200	-40 to +200	-40 to +120	-40 to +150	-100 to +250	-100 to +250	-100 to +250	-100 to +250	-100 to +250	-65 to +175	-100 to +250	-100 to +250	-100 to +250	-100 to +250	-100 to +325★	-100 to +250	-65 to +175	-100 to +200	-100 to +250	-50 to +250	-100 to +250	-100 to +250	-100 to +200	-100 to +175	-100 to +250	-100 to +250	-100 to +250	-100 to +250
Temp Coefficient %/°F	.03	.03	.03	.03	.05	.03	.03	.03	.03	.03	.03	.03	.03	.03	.03	.03	.03	.03	.03	.03	.05	.03	.03	.03	.03	.03	.03	.03	.03
Vibration (max) ±gpk	1000	1000	1000	500	1000	2000	2000	2000	1000	1000	100	500	500	1000	200	500	500	100	100	500	500	1000	1000	1000	2000	100 000	75 000	20 000	10 000
Shock (max) gpk	2000	2000	2000	1000	2000	5000★	5000★	5000	2000	10 000	5000	5000	5000★	10 000	5000	5000	5000	200	100	5000	5000	2000	2000	2000	5000	125K axial 30K trans	100K axial 30K trans	50K axial 30K trans	20K axial 20K trans
Structure (iso-compression)	upright	upright	upright	upright	upright	inverted	inverted	inverted	inverted	upright	inverted	upright	upright	upright	upright	inverted	upright	upright	upright	inverted	upright	inverted	inverted	upright	inverted	inverted	inverted	inverted	inverted
Size (hex × height) in	.281 × .48	.281 × .61	.281 × .48	.281 × .48	.23 dia★ × .30	.50 × 1.3	.50 × 1.3	.50 × 1.3	.50 × 1.5	.75 × 1.10	.562 × 1.81	.625 × 1.50	.75 × 1.32	.75 × 1.07	.75 × 1.73	.75 × 1.07	.75 × 1.32	.75 × 1.32	2.2 dia × 2.2	.75 × 2.0	1.0 dia × 2.5	1.0 cube	.625 cube	.86 cubic envelope	.50 × 1.3	.312 × .68	.312 × .68	.312 × .68	.312 × .68
Weight gm	2★	2.3★	1.9★	2★	1★	25	25	25	38	55★	39★	31	87	55	60	55	95	65	1000	87	180	130	17★	22★	25	4★	4★	4★	4★
Connector	integral[8] 10' cable	5-44 micro	integral[8] 10' cable	integral[8] 10' cable	integral[8] 6' cable	10-32	10-32	10-32	10-32	10-32	10-32	10-32	10-32	10-32	10-32	10-32	10-32	10-32	10-32	5/16-32 Con-hex	10SL★ 4P-20-0A	10-32	integral 6' cable	integral[8] 10' cable	10-32	10-32	10-32	10-32	10-32
Case Material SS	316	316	316	316	316	316	316	316	316	SS	316	SS	SS	SS	SS	SS	SS	SS	SS	SS	316	316	titanium	SS	316	17-4PH	17-4PH	17-4PH	17-4PH
Sealing	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	hermetic	hermetic	hermetic	epoxy	epoxy	hermetic	hermetic	epoxy	epoxy	hermetic★	hermetic	hermetic★	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy
Ground Isolation[6]	no	no	no	no	no	no	yes	yes	no	yes	no	yes	no	no	yes★	no	no	no	yes★	no	yes★	no	no	no	no	no	no	no	no
OPTIONAL MODELS (see specific data sheets)						**302A07** low phase		**302A05** non iso					**308B09** **308B10**	**308B11**	**308B13**		**308B31-34** 50-100-200 & 500 Hz			**308B05** base iso		**306A02** shock		**303A07** 100mV/g				**305A05**

NOTES: (★ denotes exceptional characteristic)
(1) PCB power units provide 2 to 20 mA from +18 to +28V DC supply. For 10V output, +24 to +28V DC is required. For driving cables longer than 100 ft, use 4 mA.
(2) Discharge time constant relates to low frequency response per chart and to transient events lasting up to several percent of discharge TC. For half sine events, discharge TC should be 25 times event duration. Use 484B or 484B02 Series power units to utilize the full transducer discharge TC.
(3) Standardized to tolerance indicated - nominal otherwise.
(4) Special cryogenic models operate to -440°F. Special high temperature to +325°F.
(5) Supplied with attached 10 ft ribbon wire terminating 10-32 coaxial connector
(6) Accessory isolation pads and studs are available.
(7) Calibration furnished to 5%, optional calibration available to 10% frequency range.
(8) Field repairable solder terminal connection

OPTIONS: Special sensitivities, ranges, electrical isolation, hermetic sealing, low and high pass filtering, case pressure insensitivity, lower frequency response, longer discharge TC, biased and limited output, lower noise, lower temperature electronics to -440°F or higher temperature to +325°F, conventional charge output, solder connector adaptor and thermal jackets

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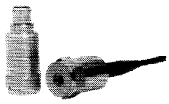
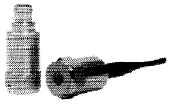
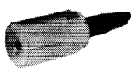
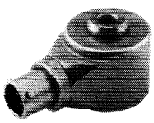
Internally Amplified Vibration Sensors

Industrial, General Purpose Accelerometers

These rugged, general purpose sensors are ideal for machinery monitoring and predictive maintenance applications. They are used on machinery monitoring systems worldwide.

	Sensitivity, mV/g	Acceleration Range, g	Frequency Response, Hz ±5%, ±10%	Temperature Range, °C	Weight, grams	Case Material	Mounting
756	100	.0007 - 50	3 - 1,000 2 - 5,000	-50 to 90	180	stainless steel	three #8 x 1/8" screws
762	10	.0015 - 500	3 - 10,000 2 - 12,000	-50 to 120	80	stainless steel	1/4-28 tapped hole
766 	100	.00055 - 50	3 - 10,000 2 - 12,000	-50 to 120	80	stainless steel	1/4-28 tapped hole
768* 	500	.00002 - 10	3 - 4,000 2 - 6,000	-50 to 120	90	stainless steel	1/4-28 tapped hole
776	100	.0002 - 40	3 - 2,500 2 - 5,000	0 to 85	32	stainless steel	1/4-28 tapped hole
793	100	.0007 - 50	3 - 5,000 2 - 7,000	-50 to 120	110	stainless steel	1/4-28 tapped hole
793L*	500	.00001 - 10	0.6 - 700 0.4 - 1,000	-50 to 120	135	stainless steel	1/4-28 tapped hole
793LS	500	.00001 - 10	0.6 - 700 0.4 - 1,000	-50 to 120	135	stainless steel	1/4-28 tapped hole
793S 	100	.0007 - 50	3 - 5,000 2 - 7,000	-50 to 120	110	stainless steel	1/4-28 tapped hole
796 	100	.0008 - 50	3 - 5,000 2 - 7,000	-50 to 120	100	stainless steel	1/4-28 captive screw

*** Ultra low noise sensor.**

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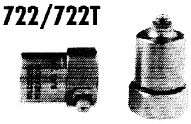
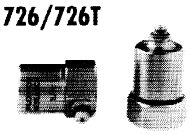
Internally Amplified Vibration Sensors

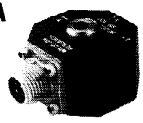
Lightweight, Compact Sensors

Lightweight, compact sensors are used in many laboratory research and light industrial applications. Their low mass makes them effective in applications where transducer mass must be minimized and high signal to noise ratio must be maintained.

	Sensitivity, mV/g	Acceleration Range, g	Frequency Response, Hz ±5%, ±10%	Temperature Range, °C	Weight, grams	Case Material	Mounting
722/722T 	10	.0002 - 500	3 - 10,000 2 - 12,000	-50 to 120	34	stainless steel	10-32 tapped hole
726/726T 	100	.0001 - 50	3 - 10,000 2 - 12,000	-50 to 120	34	stainless steel	10-32 tapped hole
728/728T*	500	.00003 - 10	3 - 4,000 2 - 6,000	-50 to 120	44	stainless steel	10-32 tapped hole
732A/732T	10	.0005 - 250	3 - 15,000 1 - 25,000	-50 to 120	15	stainless steel	10-32 tapped hole
736/736T	100	.00015 - 50	6 - 15,000 2 - 25,000	-50 to 120	15	stainless steel	10-32 tapped hole

Triaxial Accelerometers

Triaxial units contain three mutually perpendicular accelerometers in a compact housing, allowing the evaluation of three dimensional motion.

	Sensitivity, mV/g	Acceleration Range, g	Frequency Response, Hz all channels±10%	Temperature Range, °C	Weight, grams	Case Material	Mounting
733	10	.00015 - 250	2 - 4,000	-50 to 120	23	hardcoated aluminum	10-32 captive screw
755G*	3	.000005 - 1.5	6 - 4000	-50 to 80	1900	titanium	——
993A 	100	.0002 - 50	2 - 2,000	-50 to 120	90	hardcoated aluminum	¼ - 28 captive screw

*** Ultra low noise sensor.**

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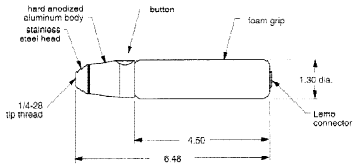
Model HA8
Hand-Held Vibration Probe - Acceleration Output

Features:
- Rugged
- Easy to use
- Shock protected with intrinsic overload protection
- Built-in electronics
- Miswiring protection

Applications:
- Predictive maintenance
- Bearing analysis
- Ship board machinery
- Heavy machinery
- Compressors



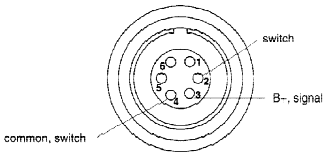
Model HA8

SPECIFICATIONS	HA8	UNITS
DYNAMIC		
Sensitivity, ±5 %, 25 °C	100	mV/g
Acceleration Range	50	g peak
Amplitude Nonlinearity	1	%
Frequency Response ±10 %:		
screwed down using mounting stud	2 - 2000	Hz
through probe tip	5 - 500	Hz
ELECTRICAL		
Power Requirement, voltage source	18 - 30	VDC
current regulating diode	2 - 10	mA
Electrical Noise, equiv. g, nom.:		
Broadband, 2.5 Hz to 25 kHz	100	µg
Spectral 10 Hz	10	µg/√Hz
100 Hz	5	µg/√Hz
1000 Hz	3	µg/√Hz
Output Impedance, max.	100	Ω
Bias Output Voltage, nom.	10	VDC
Grounding	case isolated, internally shielded	
ENVIRONMENTAL		
Vibration Limit	500	g peak
Shock Limit	100	g peak
Electromagnetic Sensitivity, equiv. g.		µg/gauss
PHYSICAL		
Weight	150	grams
Case Material	stainless steel head, hard anodized aluminum body	
Grip Material	black neoprene foam	
Output Connector (socket)	Lemo EGG1B306CNL	
Mating Connector (pins)	Lemo FGG1B306CNADA62 (R6LM)	

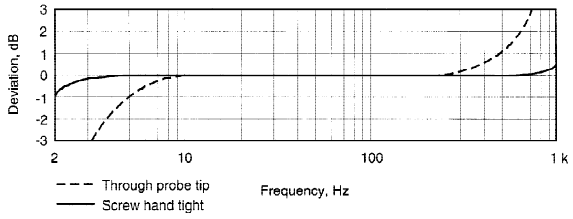
OPTIONS: Customer specified sensitivity, connector.

ACCESSORIES AVAILABLE: Cabling, power supplies, amplifiers, signal conditioners, probe tips.

Connector Pinout



TYPICAL FREQUENCY RESPONSE



Due to continued research and product development, Wilcoxon Research reserves the right to amend this specification without notice.

Wilcoxon Research, 2096 Gaither Road, Rockville, Maryland 20850 USA • 1-800-VIB-SENS or (301) 330-8811 • Fax: (301) 330-8873

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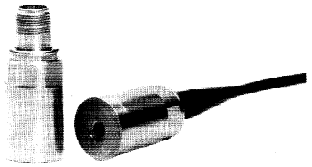
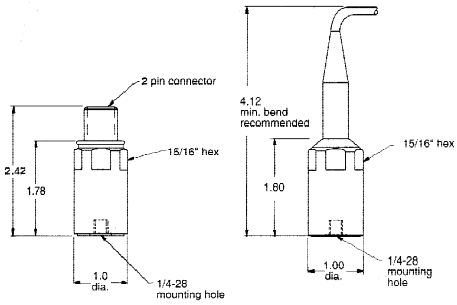
Model 793L
Low Frequency Piezoelectric Accelerometer



Features:
- Intrinsic safety certified option
- High sensitivity
- Ultra low-noise electronics for clear signals at very low vibration levels
- Low frequency capable
- Filtered to eliminate high frequencies
- Hermetic seal
- ESD protection
- Miswiring protection

Applications:
- Machinery monitoring
- Rotating machinery
- Low vibration level measurements
- Laboratory research



Model 793L **Model 793LS**

SPECIFICATIONS	793L	UNITS
DYNAMIC		
Sensitivity, ±5 %, 25 °C	500	mV/g
Acceleration Range[1]	10	g peak
Amplitude Nonlinearity	1	%
Frequency Response, nom.:		
±5 %	0.6 - 700	Hz
±10 %	0.4 - 1000	Hz
±3 dB	0.2 - 2300	Hz
Resonance Frequency, mounted, nom.	15	kHz
Transverse Sensitivity, max.	7	% of axial
Temperature Response	see graph	
ELECTRICAL		
Power Requirement, voltage source	18 - 30	VDC
current regulating diode[1, 2]	2 - 10	mA
Electrical Noise, equiv. g, nom.:		
Broadband, 2.5 Hz to 25 kHz	10	µg
Spectral 2 Hz	1.5	µg/√Hz
10 Hz	0.4	µg/√Hz
100 Hz	0.2	µg/√Hz
Output Impedance, max.	100	Ω
Bias Output Voltage, nom.	10	VDC
Grounding	case isolated, internally shielded	
ENVIRONMENTAL		
Temperature Range	-50 to 120	°C
Vibration Limit	250	g peak
Shock Limit	2500	g peak
Electromagnetic Sensitivity, equiv. g.	5	µg/gauss
Humidity Limit	100	% relative
Base Strain Sensitivity	0.001	g/µstrain
PHYSICAL		
Weight	135	grams
Case Material	stainless steel	
Mounting	¼-28 tapped hole	
Output Connector	MIL-C-5015 2-pin	
Cabling[3]: Mating Connector	Amphenol 97-3106A-10SL-4S (R6)	
Recommended Cable	J5, coaxial, polyurethane jacket, 30 pF/ft	

NOTES: [1] Signal distortion can occur when measuring high vibration levels, especially with long cables. A 24-30 VDC powering source is recommended for minimizing distortion.
[2] A maximum current of 6 mA is recommended for operating temperatures in excess of 100 °C.
[3] Cabling for "S" option model is J61, coaxial, polyurethane jacket, 30 pF/ft.

Connections
Pin A ..... Signal, power
Pin B ..... Common

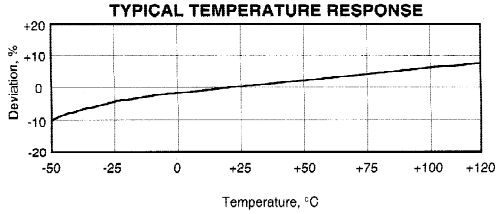
OPTIONS: Intrinsic safety "E" (793LE) certified by Factory Mutual for use in the following environments: Classes I, II, III, Div. 1, Groups A, B, C, D, E, F, G, 10 ft. sealed integral cable "S", temperature sensor "T", customer specified sensitivity, filtering, cable length.

ACCESSORIES SUPPLIED: SF6 mounting stud.

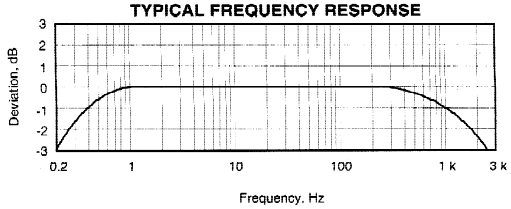
FM APPROVED

ACCESSORIES AVAILABLE: R62-J5-XX cable assembly, magnetic mounting bases, cementing studs, power supplies, amplifiers, signal conditioners.

TYPICAL TEMPERATURE RESPONSE



TYPICAL FREQUENCY RESPONSE



Due to continued research and product development, Wilcoxon Research reserves the right to amend this specification without notice.

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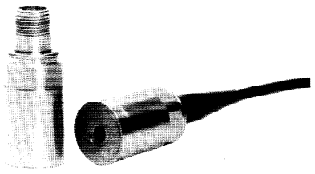
Model 793
Industrial Piezoelectric Accelerometer

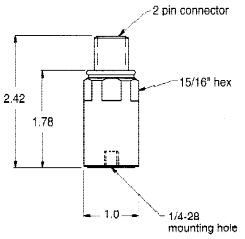


Features:

- Intrinsic safety certified option
- Rugged, general purpose
- Corrosion-resistant
- Priced economically for single and OEM quantities
- Hermetic seal
- Standardized sensitivity
- Ground isolated
- ESD protection
- Miswiring protection

Applications:

- Engine monitoring
- Machinery monitoring
- Ship board machinery
- Shock & vibration
- Predictive maintenance
- Shock pulse measurements



Model 793

SPECIFICATIONS		793	UNITS
DYNAMIC			
Sensitivity, ±5 %, 25°C		100	mV/g
Acceleration Range[1]		50	g peak
Amplitude Nonlinearity		1	%
Frequency Response:			
	±5 %	3 - 5,000	Hz
	±10 %	2 - 7,000	Hz
	±3 dB	1 - 15,000	Hz
Resonance Frequency, mounted, nom.		26	kHz
Transverse Sensitivity, max.		5	% of axial
Temperature Response		see graph	
ELECTRICAL			
Power Requirement:			
	voltage source	18 - 30	VDC
	current regulating diode[1, 2]	2 - 10	mA
Electrical Noise, equiv. g, nom.:			
Broadband, 2.5 Hz to 25 kHz		700	µg
Spectral	10 Hz	10	µg/√Hz
	100 Hz	5	µg/√Hz
	1000 Hz	5	µg/√Hz
Output Impedance, max.		100	Ω
Bias Output Voltage, nom.		12	VDC
Grounding		case isolated, internally shielded	
ENVIRONMENTAL			
Temperature Range		-50 to 120	°C
Vibration Limit		500	g peak
Shock Limit		5000	g peak
Electromagnetic Sensitivity, equiv. g.		15	µg/gauss
Humidity Limit		100	% relative
Base Strain Sensitivity		0.004	g/µstrain
PHYSICAL			
Weight		110	grams
Case Material		stainless steel	
Mounting		¼ - 28 tapped hole	
Output Connector		MIL-C-5015 2-pin	
Cabling:	Mating connector	Amphenol 97-3106A-10SL-4S (R6)	
	Recommended Cable	J5, coaxial, polyurethane jacket, 30 pF/ft	

NOTES: [1] Signal distortion can occur when measuring high vibration levels, especially with long cables. A 24-30 VDC powering source is recommended for minimizing distortion. Current regulating diode must be at least 2 mA plus 1 mA per 1000 pF of cable capacitance.

[2] A maximum current of 6 mA is recommended for operating temperatures in excess of 100 °C.

Connections
Pin A ..... Signal, power
Pin B ..... Common

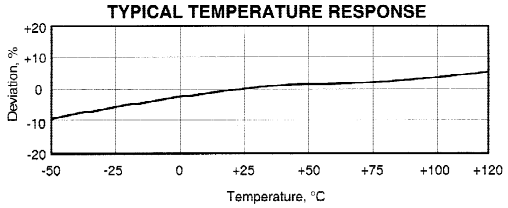
OPTIONS: Intrinsic safety "E" (793E) certified by Factory Mutual for use in the following environments: Classes I, II, III, Div. I, Groups A, B, C, D, E, F, G, 10 ft. sealed integral cable (see 793S/SG), temperature sensor "T", ultra low noise "U", customer specified filtering, sensitivity, cable length.

ACCESSORIES SUPPLIED: SF6 mounting stud.

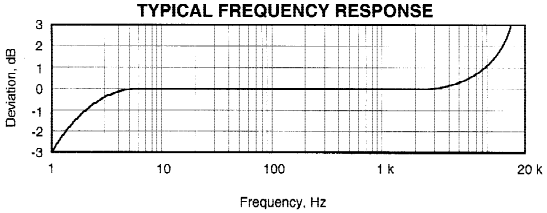
FM APPROVED

ACCESSORIES AVAILABLE: Magnetic mounting bases, cementing studs, cable assembly R62-J5-XX, power supplies, amplifiers, signal conditioners.

TYPICAL TEMPERATURE RESPONSE



TYPICAL FREQUENCY RESPONSE



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CHAPTER 3

PRINCIPLES OF DIGITAL DATA ACQUISITION AND FFT PROCESSING FOR SPECTRAL ANALYSIS

3.0 INTRODUCTION

In the previous chapter, one of the highlights was the FFT Programmable Data Collector. It is the current state-of-the-art choice for predictive maintenance programs to collect condition monitoring data. In addition, many of them have a built-in "Analysis Mode" which allows them to be used as an analyzer in the field. Since both data collectors and real time analyzers are also dynamic signal analyzers, it is important that the analyst fully comprehend the principles and techniques involved in digital data acquisition and FFT processing.

In this chapter, the basic properties of the Fast Fourier Transform will be presented. What happens when a finite, sampled record is transformed from the time domain will be considered to show the need to use filters. It will be shown that the length of the time record determines how closely the "filters" (or lines) will be spaced in the frequency domain, while the number of samples determines the number of "filters" in the frequency domain.

This chapter will highlight some undesirable characteristics, such as aliasing and leakage, and deal with solutions to these problems such as analog and digital filters and windowing. Some enhancement capabilities to make high resolution measurements, like "zoom" and averaging will likewise be featured.

Perhaps the highlight of this chapter will be the section which deals with the effects that the number of FFT lines and frequency span chosen have on the accuracy of the frequency. This is due to the fact that the more precise the displayed frequency, the more reliable will be the diagnosis of the source causing this frequency (a group of frequencies) to appear. Not only will this allow the analyst to more reliably diagnose the problem source, but also the problem severity (if a problem actually exists). Also, instruction will be given on how to improve frequency and amplitude accuracy by interpolation methods, and even choosing an instrument with sufficient dynamic range.

3.1 FFT PROPERTIES

The Fast Fourier Transform (FFT) is an algorithm for transforming the signal from the time domain to the frequency domain. However, the signal cannot be transformed to the frequency domain in a continuous manner. It must first be sampled and digitized. This means that samples from the time domain are digitized to samples in the frequency domain, as shown in Figure 1. Because of sampling, there is no longer an exact representation in either domain. However, a sampled representation can be closer to the ideal if the samples are placed closer together. Later in this section, the sampling spacing which is necessary to guarantee accurate results will be considered.

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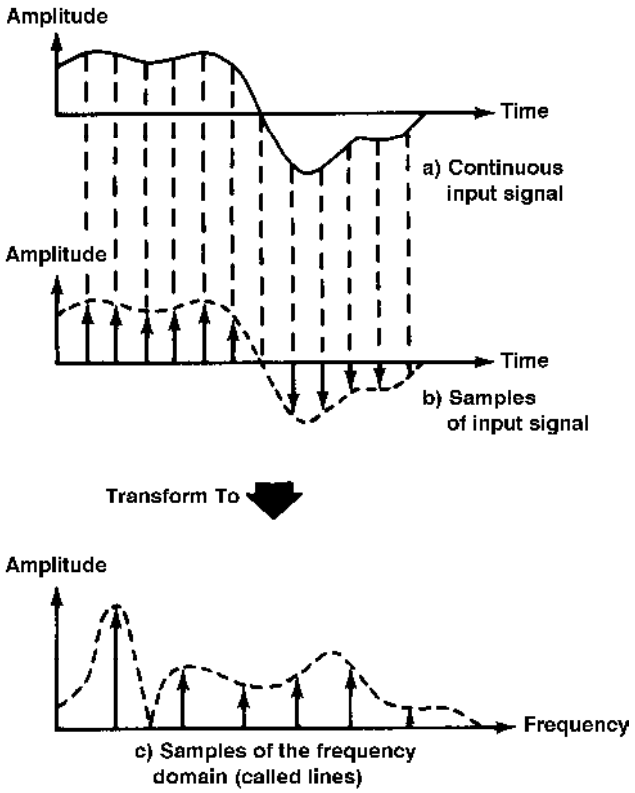

FIGURE 1
THE FFT SAMPLES IN BOTH THE TIME AND FREQUENCY DOMAINS (Ref. 1)

A time record is defined to be finite number of consecutive, equally-spaced samples of the input signal. Because it makes the transform algorithm simpler and much faster, the finite number of samples is restricted to a multiple of 2. Microprocessors work in powers of 2, called binary numbers. Thus, 1024 intervals or equally-spaced samples equal 2^{10} intervals (if an A/D instrument was specifically designed to do this, it would be referred to as a "10-BIT A/D INSTRUMENT").

This digitized time record is transformed as a complete block into a complete block of frequency lines. All the samples (of the input signal) of the time record are needed to compute each and every line in the frequency domain. This does not mean a single time domain sample transforms to exactly one frequency domain line. An explanation on this will be given in the next section.

Because the FFT transforms the entire time record as a total, there are no valid frequency domain results until the complete time record has been gathered. Once gathered, however, the oldest sample can be discarded, all the samples shifted in the time record, and a new sample added to the end of the time record as in Figure 2. Thus, once the time record is filled initially, there will be a new time record at every time domain sample. The point is, that after the initial time span, there will be rapid changes in the spectra. Although this is not practical, it could be utilized.

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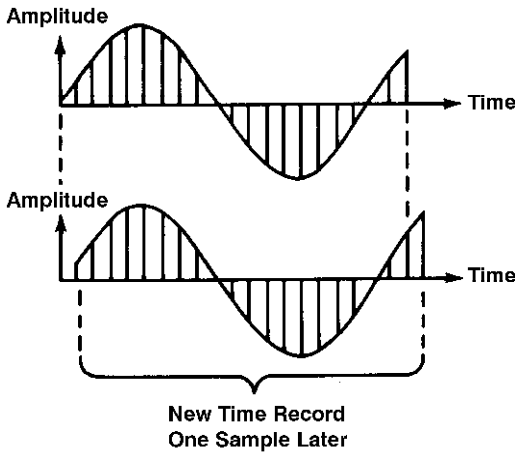

FIGURE 2
A NEW TIME RECORD ADDED EVERY SAMPLE AFTER THE TIME RECORD IS FILLED (Ref. 1)

3.11 HOW MANY SPECTRAL LINES ARE THERE?

The FFT algorithm is a "complex-valued" operation; that is, it produces a "real" and an "imaginary" result, and the components of the frequency domain will appear at both positive and negative frequencies. The complex plane is 0 (inclusive) to 2π. The positive vector lies in the upper half of the complex plane. So a positive frequency is located between 0 and π. The negative vector lies in the lower half of the complex plane. Thus, a negative frequency is located between π and 2π (remember there are 2π radians per revolution of a rotating shaft).

For real-valued signals, it can be shown that the components with positive frequencies are mirror images of their corresponding components with negative frequencies. Therefore, it is common practice to discard the redundant, negative frequency values when graphing the amplitude of the FFT of a real-valued signal. Since only half of the information returned from the FFT algorithm is displayed, component amplitudes are multiplied by two before being displayed (except for the DC component, which is at zero frequency on the real axis).

In summary, each of these components in the frequency domain is complex-valued; that is, they each have both amplitude and phase. Thus, the FFT transforms a finite number of equally-spaced samples from the time domain to only half as many lines in the frequency domain (see Equation 1). The reason for this is the same as stated above - each frequency line actually contains two pieces of information - amplitude and phase.

Thus, for a real-valued signal (the numbers are not complex), 800 data points would be required to compute a 400 line spectrum; 1600 data points for an 800 line spectrum, etc. However, some of the high frequency data is "discarded" to compensate for the roll-off of the anti-aliasing filters. In the "real world", 1024 data points are actually required for a 400-line transform; 2048 points for an 800-line transform, etc.

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3.12 WHAT IS THE SPACING OF THE LINES?

The lowest frequency that can be resolved must be based on the length of the time record. If the period (T) of the input signal is longer than the time record, there is no way of determining the period. Therefore, the lowest frequency line of the FFT must occur at a frequency equal to the reciprocal of the time record length.

$F = \frac{1}{T}$ where: F = Frequency (Hz or CPM)
T = Period (sec or min)

In addition, there is a frequency line at zero Hz-DC. This is merely the average of the input over the time record. Although it has no practical value, it does help to establish that the spacing between these two lines (and hence every line) is the reciprocal of the time record. Thus, the entire spectral display is made up of many individual vertical lines (or "bins") located adjacent to one another along the frequency axis. Only bins with amplitude information in them will display an FFT peak.

3.13 WHAT IS THE FREQUENCY RANGE OF THE FFT?

The highest frequency which can be measured is:

$$F_{MAX} = \frac{SAMPLE\ SIZE}{2} \times \frac{1}{PERIOD\ OF\ TIME\ RECORD} \qquad \text{Eqn. 1}$$

because there are only 1/2 as many lines in the frequency domain, spaced by the reciprocal of the time record starting at zero Hz. The usefulness of this frequency range can be limited by a problem called "aliasing" which will be discussed in a later section.

The number of time samples (sample size) is fixed by the implementation of the FFT algorithm. Therefore, the period of the time record (sec/cycle) must be varied to change F_{MAX} (cycles/sec). To do this, the sample rate must be varied so that there always is the chosen, fixed number of time samples in the variable time record period. To cover high frequencies, the time record period must be shorter so that sampling is very fast.

3.2 SAMPLING AND DIGITIZING

Recall that the input is a continuous analog voltage coming from the accelerometer and is proportional to the acceleration. Since the FFT requires digitized samples of the input for its digital calculations, a "sampler" and an "analog to digital converter" (A/D) needs to be added to the FFT processor to create a spectrum analyzer.

For the analyzer to have the high accuracy needed, the sampler and the A/D converter must be quite good. The A/D converter must have high resolution and linearity. For 70 dB of dynamic range, the A/D converter must have at least 12 bits of resolution (a "12-bit A/D instrument has 4096 equally-spaced samples or intervals in the display, equal to 2^{12} intervals). Typically, the A/D converter must be able to acquire at least a hundred thousand readings per second.

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3.3 ALIASING

The reason an FFT spectrum analyzer needs so many samples per second is to avoid "aliasing". Aliasing is not always bad. It is called "mixing" or "heterodyning" in analog electronics, and is commonly used for tuning household radios and televisions.

3.31 ALIASING IN THE FREQUENCY DOMAIN

It is easy to see that a sampling frequency that is exactly twice the input frequency would not always be enough in the time domain. If the sampling rate is low (<2), the alias products will fall in the frequency range of the input ("creating" fictitious frequencies) and no amount of filtering will be able to remove them from the signal. However, as shown in Figure 3, if we sample at greater than twice the highest frequency of our input, the alias products will not fall within the frequency range of our interest. Therefore, a filter (or the FFT processor which acts like a filter) which is placed after the sampler will remove the alias products, but pass the desired input signals.

Two signals are said to alias if the difference of their frequencies falls within the frequency range of interest. This difference frequency is always generated in the process of sampling.

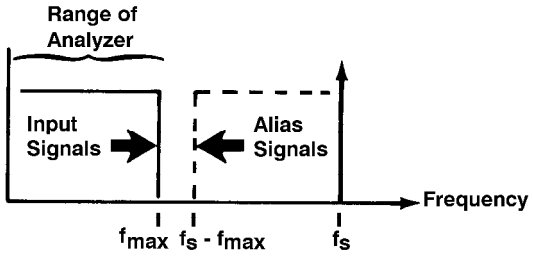

FIGURE 3
A FREQUENCY DOMAIN VIEW OF HOW TO AVOID ALIASING (Ref. 1)

3.32 THE NEED FOR AN ANTI-ALIAS FILTER

The only way to be certain that the input frequency range is limited is to add a low-pass filter before the sampler and the A/D converter. Such a filter is called an anti-alias filter. The gradual roll-off area of the filter is known as the transition band. Large input signals are not well attenuated in the transition band; they can still alias. To avoid this, the sampling "frequency" is raised to twice the highest frequency of the transition band. Typically, this means that to do this the sample rate actually is now 2.5 to 4 times the maximum desired input frequency.

3.33 THE NEED FOR MORE THAN ONE ANTI-ALIAS FILTER

To reduce the frequency span, the sample rate and the anti-alias filter frequency must be reduced by the same amount. To cover a wide range of frequencies, there needs to be many anti-alias filters (at least 12) for each channel with this setup. Unfortunately, they are very expensive, particularly the lowest frequency filters. Also, a different filter has to be utilized every time the sample rate of the A/D converter is changed. These are analog filters.

3.34 DIGITAL FILTERING

A digital filter filters the input signal after it has been sampled and digitized. The A/D sample rate is set to the rate needed for the highest frequency span. Rather than changing to a different anti-alias filter each time, a single analog anti-alias filter is used. Then, a digital filter is added after the A/D converter. It not only filters the digital representation of the signal to the desired frequency span, but it also reduces the sample rate to the needed rate of the frequency span.

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3.35 FORMULAS USED TO CALCULATE t_{MAX} AND F_{MAX}

Nyquist's Sampling Theorem, briefly stated, is sampling must be done at a frequency rate of at least twice the highest frequency component of interest in order to not lose any information contained in the sampled signal. Thus, to satisfy Nyquist's Theorem, select a sampling rate that is slightly higher than twice F_{MAX} (2.56 X F_{MAX} is typical). The following are conversion formulas to calculate the time required to gather data if the F_{MAX} is known and vice versa.

$$t_{MAX} = \frac{(60)(\#FFT\ Lines)}{FREQ.SPAN} = \frac{(60)(SAMPLE\ SIZE)}{(2.56)(FREQ.SPAN)} \qquad \text{Eqn. 2}$$

where:

t_{MAX}	=	Total Sampling Period setting how often amplitude is measured and stored (sec).
SAMPLE SIZE	=	Number of Analog to Digital Conversions to be used to Construct the Time Waveform (Samples - most often 1024 Samples to provide a 400 line FFT).
FREQ. SPAN	=	Frequency Span from F_{MIN} to F_{MAX} (CPM); which often is from 0 - F_{MAX}.

$$F_{MAX} = \frac{(60)(\#FFT\ Lines)}{t_{MAX}} = \frac{(60)(SAMPLE\ SIZE)}{(2.56)(t_{MAX})} \qquad \text{Eqn. 3}$$

where:

t_{MAX}	=	Total Sampling Period setting how often amplitude is measured and stored (sec).
SAMPLE SIZE	=	Number of Analog to Digital Conversions to be used to Construct the Time Waveform (Samples - most often 1024 Samples to provide a 400 line FFT).
F_{MAX}	=	Maximum Spectral Frequency or Frequency Span (CPM)

If this condition is violated (by not using an anti-aliasing filter), higher frequencies will be represented as lower frequencies in the sampled waveform. An example of this phenomenon may be seen in old western movies in which the wagon wheels appear to rotate backwards; this is because the film speed (sampling rate) was too slow compared to the speed of the wagon wheels. Vibration analysts can observe this same effect if they vary the frequency of a strobe light that is flashing on a part that is rotating at constant speed.

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3.4 WINDOW SELECTION

3.41 THE NEED FOR WINDOWING

If a sine wave is periodic in the time record, its frequency spectrum would be a single FFT line if no windowing was used. However, if a sine wave is not periodic in the time record, its power will be spread throughout the spectrum. This smearing of energy throughout the frequency spectrum domain is known as "leakage". Energy leaks out of one resolution line of the FFT into all other lines as shown in Figure 4.

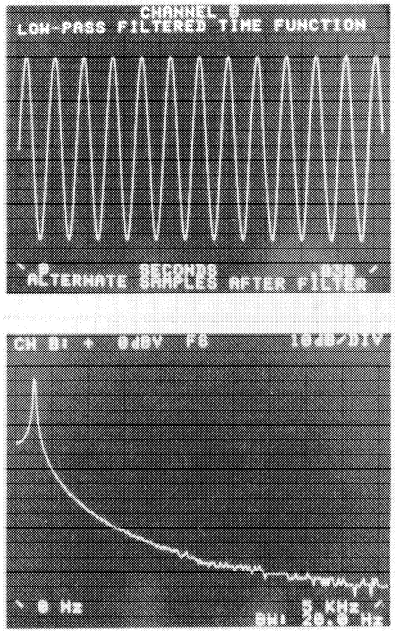

FIGURE 4
LEAKAGE OCCURS WHEN SINE WAVES ARE NOT PERIODIC IN THE TIME RECORD, AND AFFECTS THE REMAINDER OF THE SPECTRUM (Ref. 1)

It is obvious here that the leakage problem can entirely mask small signals located close to the sine wave(s). The solution to this problem is known as "windowing".

Notice that most of the problem seems to be at the edges of the time records; the center portion is a good sine wave. If the FFT could be made to ignore the ends and concentrate on the middle of the time record, the spectrum in the frequency domain should more closely produce a single peak. Windowing accomplishes this task - it forces the sampled data at the beginning and at the end of the sampling period to be equal to zero, thereby minimizing leakage.

3.42 WHAT IS WINDOWING?

If the time record was multiplied by a function that is zero at the ends of the time record and large in the middle, the FFT could be concentrated on the center portion of time record. One such function is shown in Figure 5A(c). Such functions are called "window functions" because they force us to view data through a narrow window. Figure 5A(b) shows the discontinuity in the time record. By applying a (HANNING) window, as in Figure 5A(c), the discontinuity is "filled in" so that the sample signal appears to be continuous, as shown in Figure 5A(d). Thus, the sampled data is forced to start and end at zero. Figure 5B is a simplified illustration of a one-cycle waveform in which the time record is not periodic. With Figure 5B(b) a (HANNING) window applied, the non-periodic waveform is forced to "appear" to be "periodic".

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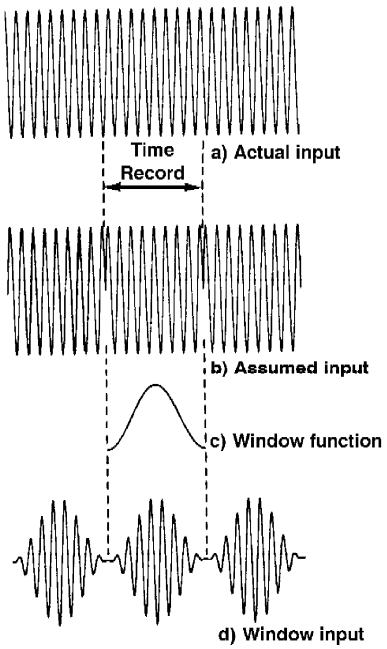

FIGURE 5A
THE EFFECT OF WINDOWING IN THE TIME DOMAIN (Ref. 1)

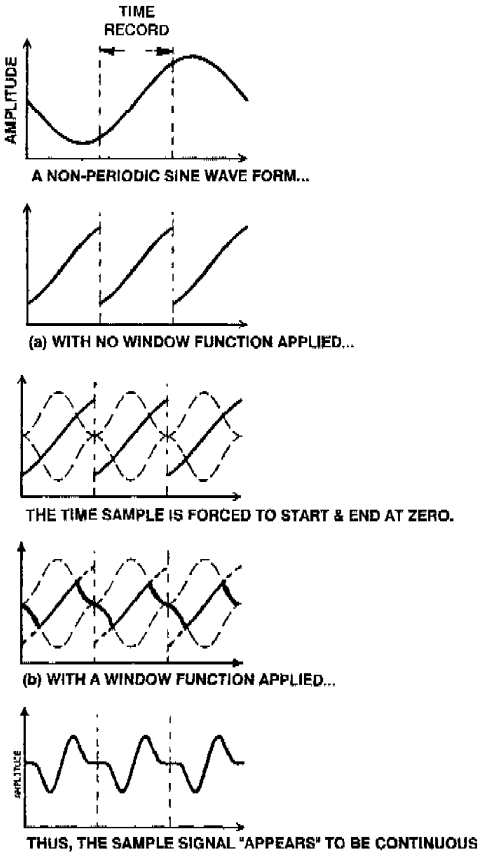
TIME
RECORD
AMPLITUDE
A NON-PERIODIC SINE WAVE FORM...
(a) WITH NO WINDOW FUNCTION APPLIED...
THE TIME SAMPLE IS FORCED TO START & END AT ZERO.
(b) WITH A WINDOW FUNCTION APPLIED...
THUS, THE SAMPLE SIGNAL "APPEARS" TO BE CONTINUOUS

FIGURE 5B
A SIMPLIFIED ILLUSTRATION TO SHOW HOW WINDOWING FORCES A NON-PERIODIC WAVEFORM TO "APPEAR" TO BE PERIODIC (Ref. 3)

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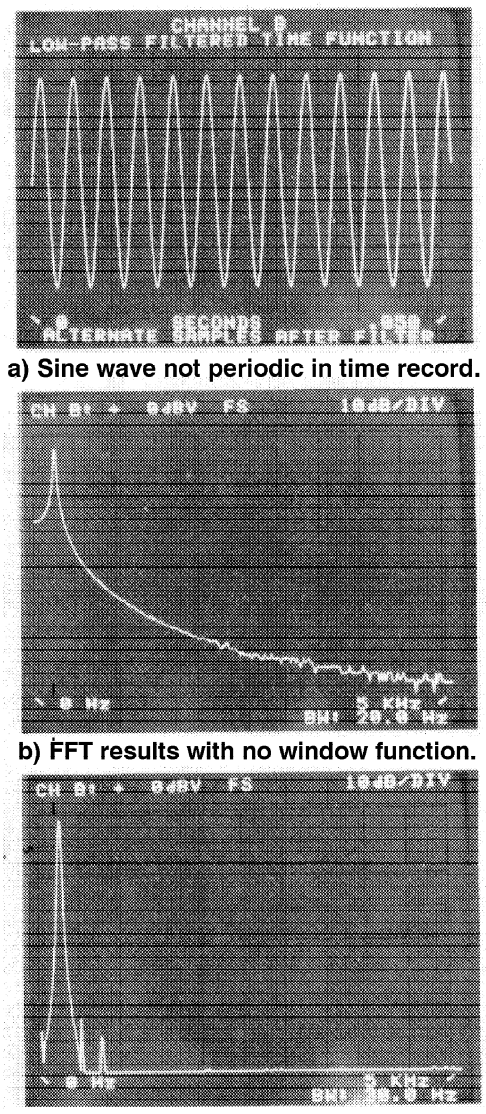
However, it is important to realize the input data has been tampered with by windowing and perfect results cannot be expected. The FFT assumes the input looks like figure 5A(d), something like an amplitude-modulated sine wave. This will produce a frequency spectrum that will be closer to a single line peak than one would be from Figure 5A(b), but it is still not 100% correct. Although Figure 6 shows a vast improvement from windowing data which is not periodic in the time record, it may still not be correct in that spectral components (or sidebands) might be "produced", but none truly exist in reality. Figure 7 demonstrates that the windowed data does not have as narrow a spectrum as an unwindowed function which is periodic in the time period.



a) Sine wave not periodic in time record.

b) FFT results with no window function.

c) FFT results with a window function.

FIGURE 6
LEAKAGE REDUCTION WITH WINDOWING (Ref. 1)

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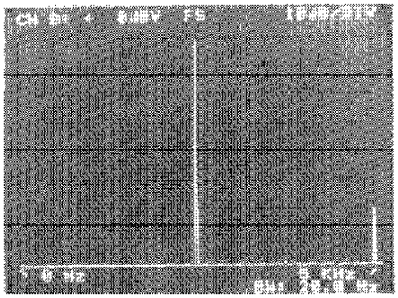
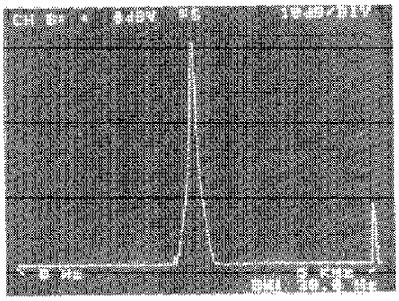


a) Leakage free measurement - input periodic in time record



b) Windowed measurement - input not periodic in time record

FIGURE 7
WINDOWING REDUCES LEAKAGE, BUT DOES NOT ELIMINATE IT (Ref. 1)

3.43 THE HANNING WINDOW

The most commonly used window in vibration analysis is the Hanning window. In fact, the window used in Figure 5 was a Hanning window. The Hanning window does a good job with sine waves, both periodic and non-periodic. However, even with the Hanning window, some leakage will be present when the signal in the time record is not periodic.

Recall that the FFT acts like a set of parallel filters. Figure 8 shows the shape of those filters when Hanning windows are used. Notice that the Hanning function gives the filter a very rounded top. If a component of the input signal is centered in the filter, its amplitude will be measured accurately. In fact, it will be periodic in the time record. On the other hand, the filter shape will attenuate the component by up to 1.5 dB (-16%) when it falls midway between the filters.

To illustrate, using Figure 8 consider the following example:

Induction Motor Speed	=	1785 RPM
F_{MAX}	=	12,000 CPM
# FFT Lines	=	400 lines (Hanning Window)
Freq. Resolution	=	30 CPM/line (1785 RPM will fall midway between 1770 and 1800 CPM bins meaning its amplitude reading is 16% lower than actual amplitude).
If F_{MAX}	=	6,000 CPM, 1785 RPM would fall in the middle of a bin, meaning there would be no reduction in the amplitude reading.

Thus, with the Hanning Window, amplitude measurements are always +0% to -16% of the actual.

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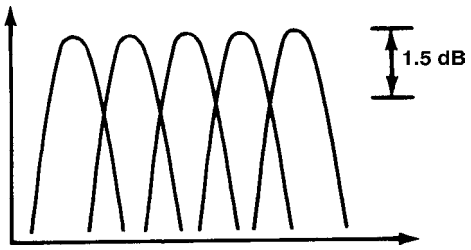


FIGURE 8
HANNING WINDOW PASSBAND SHAPES (Ref. 1)

That is, while the Hanning window provides improved frequency resolution, it sacrifices amplitude accuracy. Therefore, Hanning windows will lose information from transient events such as the impact test shown in Figure 9a. If the transient input signal is multiplied by the Hanning window function in Figure 9b, it would produce the highly distorted signal shown in Figure 9c. The frequency spectrum of an actual transient with and without the Hanning window is shown in Figure 10. The Hanning window has taken the transient, which naturally has energy spread widely through the frequency domain and made it look more like a sine wave.

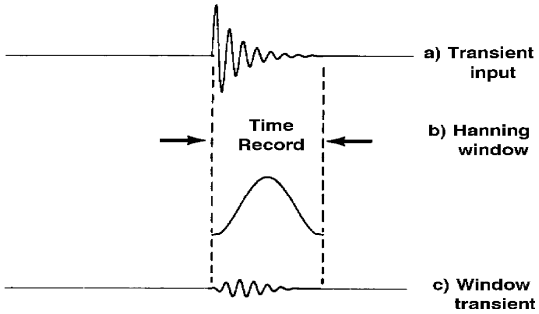
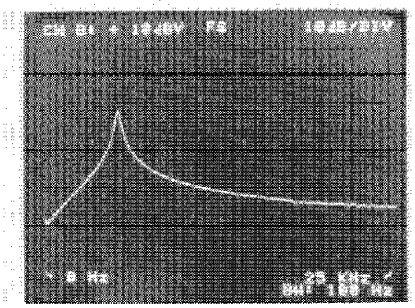


FIGURE 9
A HANNING WINDOW LOSES INFORMATION FROM TRANSIENT EVENTS (Ref. 1)


a) Unwindowed transient

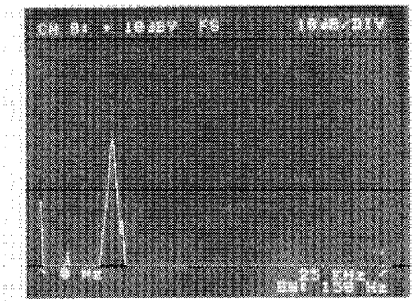

b) Hanning windowed transient

FIGURE 10
SPECTRA OF TRANSIENTS WITH AND WITHOUT A HANNING WINDOW (Ref. 1)

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3.44 THE UNIFORM (RECTANGULAR) WINDOW

Since Hanning windows do not work well with transient vibrations, a uniform window (no window) could be used because it weights all of the time record uniformly. In the case of a transient, the analyzer can be set up so that there is no need for windowing the input because the transient occurs completely within the time record. This is called a "self-windowing function". It generates no leakage in FFT and so needs no window. Uniform windows are very useful for impact natural frequency tests. On the other hand, if a uniform window is used on a non-periodic sine wave, it will not eliminate leakage or wide skirts in the spectrum. In addition, amplitude variation for a uniform window can be up to 36%.

3.45 THE FLAT TOP WINDOW

Recall that the Hanning function gives the filter a very rounded top. Although this characteristic is desirable to clearly identify frequency peaks, it is unacceptable if the signal's amplitude needs to be accurately measured. The solution is to provide a window function that gives the filter a flatter passband, as shown in Figure 11.

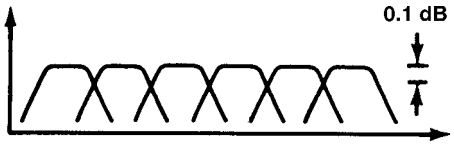

FIGURE 11
FLAT TOP PASSBAND SHAPES (Ref. 1)

The amplitude error from this window function does not exceed .1 dB (1%). The accuracy improvement does not come without its price, however. Figure 12 shows that the top of the passband has been flattened at the expense of widening the skirts of the filter. Therefore, some of the ability to resolve a small component, closely spaced to a large one, is lost.

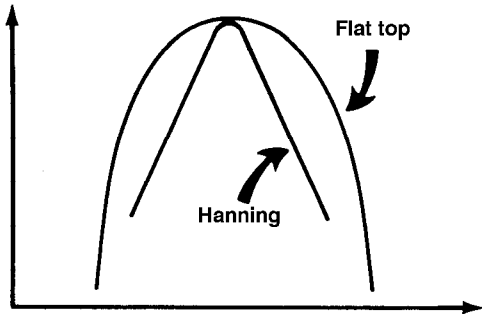

FIGURE 12
COMPARISON OF HANNING TO FLAT TOP WINDOW SHOWS REDUCED FREQUENCY RESOLUTION CAPABILITY OF THE FLAT TOP WINDOW (Ref. 1)

Many other window functions are possible but the three listed above are by far the most common for general measurements. In summary, the Flat Top window provides the best amplitude measurement, the Kaiser-Bessell window provides the best frequency separation, whereas the Hanning window provides the best compromise.

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3.5 AVERAGING

The standard technique in statistics to improve the estimates of a value is to average. Two kinds of averaging are usually available, RMS (or "power" averaging) and synchronous time averaging.

3.51 RMS (POWER) AVERAGING

RMS averaging can be thought of as amplitude averaging. Each frequency's component in a power spectrum is proportional to the square of the corresponding component in the Fourier Transform. For the purpose of emphasizing the amplitude of the FFT, the analyzer will compute the power spectrum first; then take the square root of each component.

Note two practical consequences: the power spectrum requires less calculation time than the Fourier Transform because the square root calculations are not performed. Thus, any contrast between the high and low amplitudes is accentuated because the quantities are squared. Naturally, the Power Spectrum does not provide any phase information at all; only the differences in amplitude are emphasized. What we are trying to determine is, the average (or RMS) amplitude of the signal.

This averaging technique is very valuable for determining the average (RMS) power in any of the filters of an analyzer. The more averages that are taken, the better the estimate of the power level, making it more and more accurate.

If a small signal is measured in the presence of noise, the signal-to-noise ratio will not be improved with RMS averaging. RMS averaging does not "filter out" the noise; it includes the noise. Thus, the accuracy of the total power (signal-plus-noise) will be improved with RMS averaging, but not the signal-to-noise ratio. It does nothing to reduce unwanted background noise.

RMS averaging should be used when spectral-component-amplitude levels vary significantly. The result of an RMS average of successive FFT's is an improved estimate of the mean level of the amplitude. It causes varying amplitudes to converge to their mean value.

3.52 LINEAR AVERAGING AND SYNCHRONOUS TIME AVERAGING

The major difference between linear and synchronous time averaging is that a "triggering device" is needed for a synchronous time average, but is not necessary for a linear average. The triggering device is used to provide a synchronizing signal; the most commonly used devices are a photocell or an electromagnetic pickup.

Unlike power averaging, linear averaging does improve the signal-to-noise ratio. In many cases, the amplitudes of fundamental and passing frequencies are fairly constant, but is accompanied by some random amplitudes and frequencies. Even in cases where the amplitudes are low, the discrete frequency amplitudes remain fairly constant, while the random signal amplitudes will fluctuate between zero and some peak value. As a result, the average amplitudes of the random signals will be significantly less than their peak values. In both cases, the amplitudes that are stable remain close to the same value, regardless of the number of averages taken. On the other hand, random or transient amplitudes tend to approach zero as more and more averages are taken. Figure 13 shows a time record buried in noise. After taking 128 averages in this case, the resulting time record and frequency spectrum shows a marked improvement in the signal-to-noise ratio.

Linear averaging is a useful method to obtain repeatable data for fault-trending in PMP's. Usually 4, 8, or 16 averages are specified in the program software. That is, if it desired to average 16 samples, the "averager" will automatically proceed to sample the data 16 times, each time

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representing 1/16 of the data present. After the 16th sample has been completed, the averaged spectrum, retained in memory, is displayed. Since the vibration sensed at each measurement point is the total vibration from the whole environment, the disadvantage of linear averaging is that it does not differentiate between the vibrations that only originate from the point and those that are only "sensed" at that point.

On the other hand, synchronous time averaging can be used to differentiate between vibration signals originating directly from the machine component of interest and eliminating those from nearby machines or any nonsynchronous components. In this case, a synchronizing signal is used to "trigger" the start of the time record. In this manner, a sample of the vibration is taken at the same moment with respect to shaft rotation each time. Therefore, the periodic part of the input where the trigger is targeted will always remain the same in each time record taken. If a series of these triggered time records are added together and divided by the number of records, N, taken, the synchronous time average can be computed.

Only harmonic vibrations from the machine (or shaft) being targeted by the trigger (photocell, strobe, laser tach, etc.) are emphasized by synchronous time averaging . The nonsynchronous frequencies will not be displayed because they are not integer multiples of the 1X RPM synchronizing signal. Thus, bearing defect frequencies, electrical line frequencies and multiples, as well as other harmonically distorted signals (even 1X RPM multiples from a nearby shaft), will be forced to zero (even if the speed difference is small like 1 or 2 RPM) if many averages are taken. This method is very useful for analyzing equipment that have many components rotating at nearly the same speed, such as a bank of fans, rolls on a paper machine, or on gears and shafts in a gearbox. After both results are transformed to the frequency domain, the running speed and harmonics are clearly seen and can be accurately measured because of the reduced noise floor (signal-to-noise ratio is enhanced).

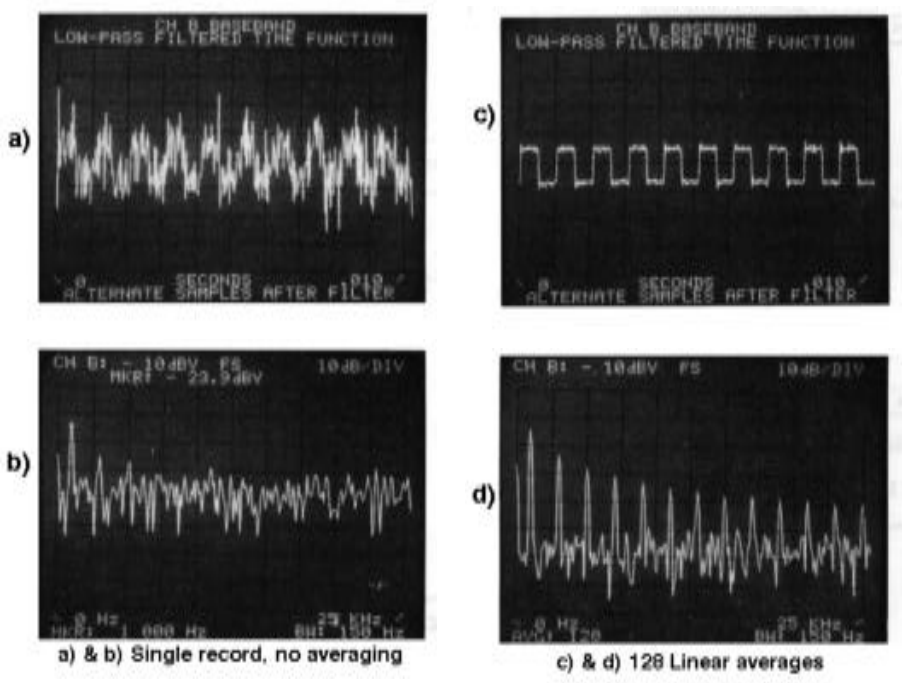

FIGURE 13
SYNCHRONOUS AVERAGING REDUCES THE NOISE FLOOR AND ALLOWS THE HARMONICS TO BE EMPHASIZED (Ref. 1)

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3.6 OVERLAP PROCESSING

Recall that the time record is not constant but is deliberately varied to change the frequency span of the analyzer. For wide frequency spans the time record is shorter. That is, first it will take a certain amount of time to capture the first time record according to the laws of physics, no matter whose analyzer one might have and whether or not they have a 32-bit DSP chip installed or not. The required time for this first block is given by Equation (2) which is repeated here for the reader:

$$t_{MAX} = \frac{(60)(\#FFT\ Lines)}{FREQ.SPAN} = \frac{(60)(SAMPLE\ SIZE)}{(2.56)(FREQ.SPAN)} \qquad \text{Eqn. 2 (Repeated)}$$

After capturing the time record, it will then require a finite time to compute the FFT (this is where the high speed processors have an effect).

This fact cannot be overlooked. It will be especially important when a low frequency analysis needs to be made. In this case, the gathering of a time record requires much longer than the FFT computation time. Thus, the FFT processor sits idle much of the time. This can be seen in Figure 14 where the time record and the FFT are represented as "blocks" of time. If, instead of waiting on an entirely new time record, the new record was "overlapped" with some of the old data, a new spectrum could be obtained as often as the FFT was computed (after capturing the first block as per Equation 2). This is "Overlap Processing" and is illustrated in Figure 15. Notice that an FFT is obtained at the end of each time record. The time (space) between each new FFT represents the time it requires the processor to display the FFT.

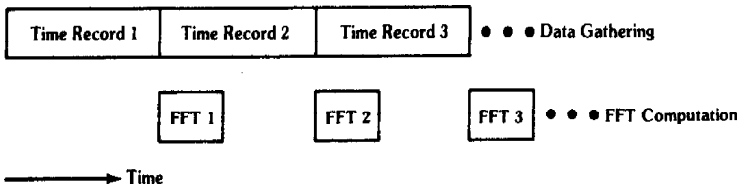

FIGURE 14
NON-OVERLAPPED PROCESSING IS PERFORMED ONLY ON COMPLETELY NEW DATA (TIME RECORDS) (Ref. 1)

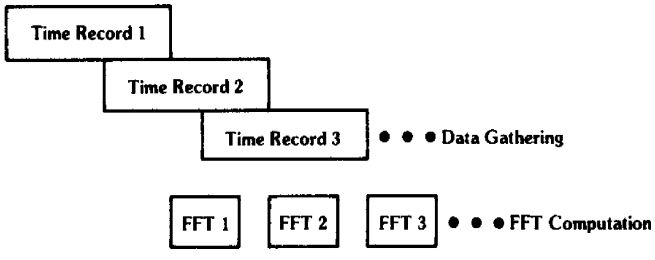

FIGURE 15
OVERLAPPED PROCESSING IS PERFORMED ON DATA THAT COMBINES OLD & NEW TIME BETWEEN FFT'S REPRESENTING PROCESSING (Ref. 1)

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In essence, a full length time record is first captured. Then portions of new time records are stored and added to portions of the old data. With an overlap of 50%, half of the new time record is taken in and combined with half of the old time record. And, since only half of the new time record is taken in, this cuts the FFT processing time down by a factor of 2. If, on the other hand, one were to ask for 75% overlap, he can cut the time by a factor of approximately 4 times, using only 25% of new information with each calculation. This can greatly speed up the measurement process, especially when capturing 8 or more averages and is acceptable for most condition monitoring measurements - as long as the vibration is periodic (that is, repeating, predictable and without pronounced transients.

Of course, as is the case with many "speed" techniques, one does not get something for nothing. Because an overlapped time record contains old data from before the device adjustment, it is not completely correct. It does indicate the direction and the amount of change, but one must wait for a full time record after the change for the new spectrum to be accurately displayed. Therefore, for measurements such as transient measurements which occur shorter than the time record, overlap processing is useless. For transients longer than the time record, the real-time bandwidth of the analyzer and spectrum recorder is usually a limitation. If it is not, overlap processing allows more spectra to be generated from the transient, usually improving the spectral data captured during the event. In any case, overlap processing can give dramatic reductions in the time required to compute averages and eliminates most of all of the time that would be wasted by the FFT processor waiting for a new time record. For most measurements that would occur during predictive maintenance vibration surveys, overlap processing can be used with a good degree of confidence. Therefore, overlap processing can be quite handy for low-frequency measurements which otherwise would require excessive periods of measurement time.

3.61 EXAMPLE OF SAMPLING TIMES WITH AND WITHOUT OVERLAP PROCESSING

The total time required for these measurements is a function of the time required to calculate the FFT from the measured time waveform as well as the time required for the transducer and data collector system to "settle". It will likely require from 2 to 3 minutes per point on low-speed machines (including settling time). Regardless of the instrument used, the equation which will indicate how long it will take to one time record is as follows:

$$DATA\ SAMPLING\ TIME = \frac{60(\#SAMPLES)(\#AVG.)}{(2.56)(FREQ.\ SPAN)} = \frac{60(\#FFT\ LINES)(\#AVG.)}{FREQ.\ SPAN} \qquad \textbf{Eqn. 4}$$

where:

DATA SAMPLING TIME	=	Total Sampling Period (setting how often amplitude is measured and stored), seconds. (Assuming no Overlap Processing is involved)
#SAMPLES	=	Number of Analog to Digital Conversions to be used to Construct the Time Waveform (1024 samples for 400 line FFT)
#AVG.	=	Number of Averages
FREQ. SPAN	=	Frequency Span (CPM) (Extending from F_{MIN} to F_{MAX} for systems allowing FFT measurement to begin at a frequency other than 0 CPM).

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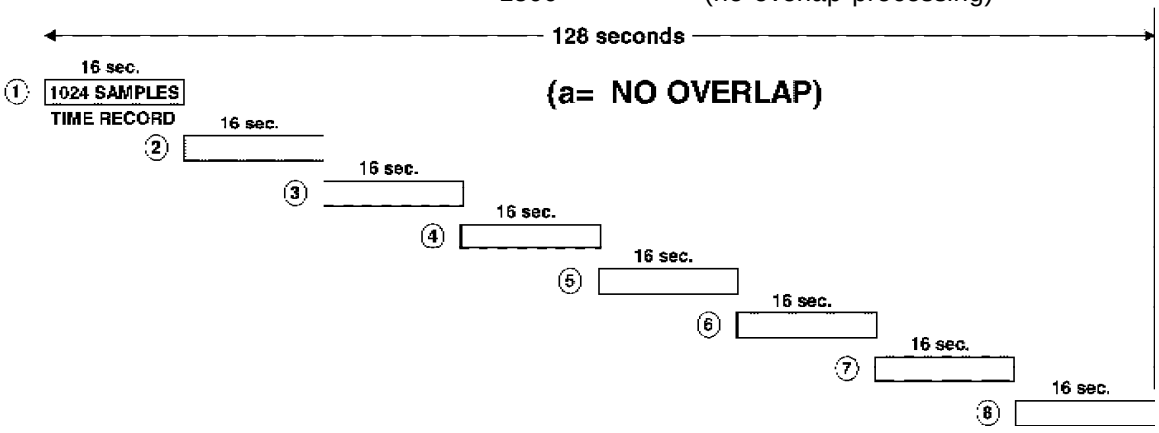
The following example will illustrate the dramatic reduction in sampling time which can be realized by invoking overlap processing:

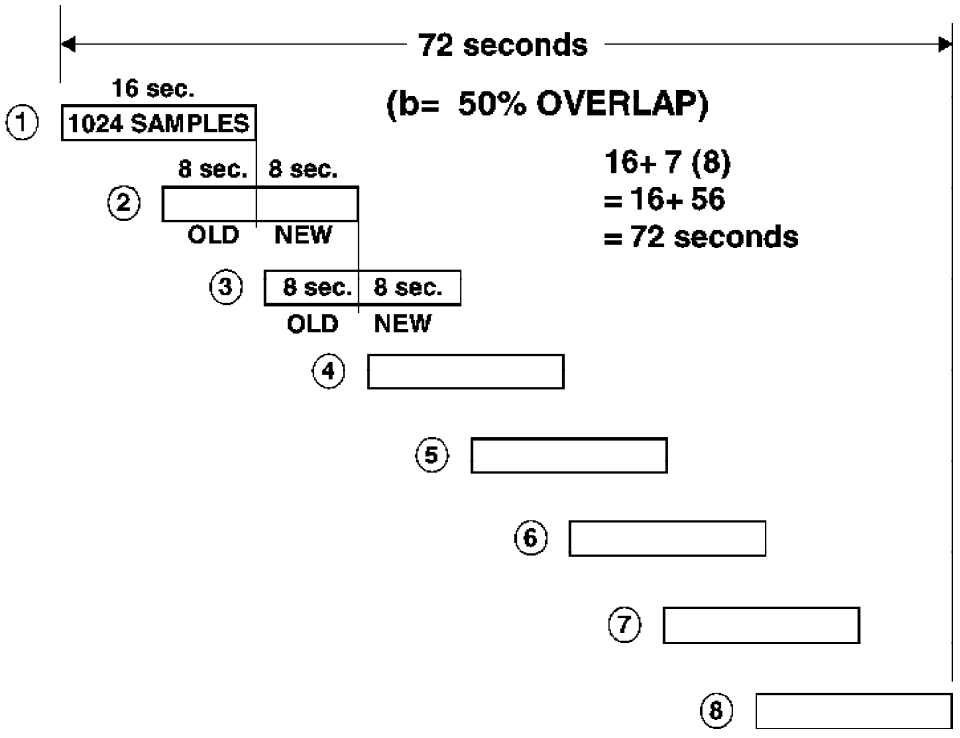
EXAMPLE COMPARISON OF REQUIRED SAMPLING TIME WITH & WITHOUT USING OVERLAP PROCESSING:

For an example, if it was necessary to capture a 400-line FFT, with 8 averages and a frequency range from 0 to 1500 CPM (0 - 25 Hz), it would require 128 seconds just to sample the data (not including transducer and data collector system settling time).

$$\text{Data Sampling Time} = \frac{(60)(400\ \text{lines})(8\ \text{ave.})}{1500} = \frac{128\ \text{seconds}}{\text{(no overlap processing)}}$$



a) No Overlap Processing - 8 averages @ 16 seconds per time record = 128 sec.



b) With 50% Overlap Processing - 4.5 averages @ 16 seconds for the first time record and 8 seconds for each subsequent time record = 72 sec.

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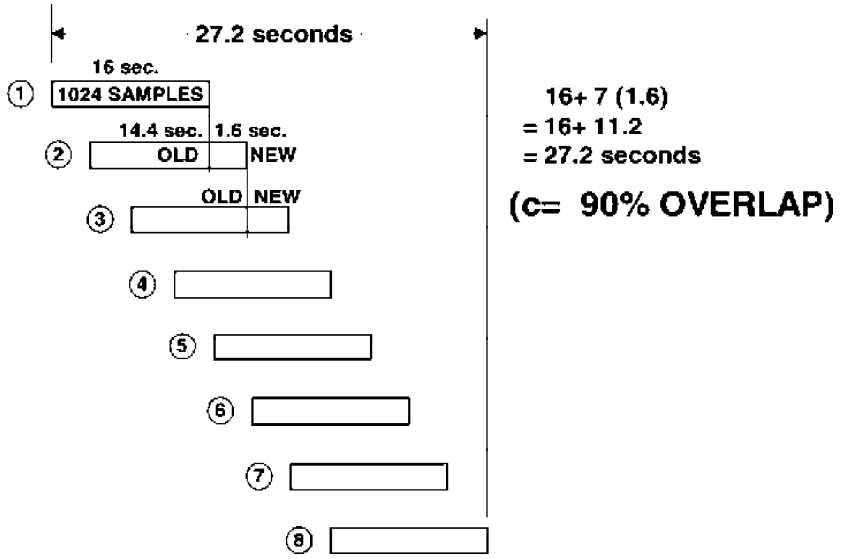

c) With 90% Overlap Processing - 1.7 averages @ 16 seconds for the first time record and 1.6 seconds for each subsequent time record = 27.2 sec.

While it is not advisable to specify 90% overlapping for condition monitoring surveys (even if the software has the capability), it becomes apparent that much sampling time is saved by specifying at least 50% overlapping. 90% overlap (or above) can be used when the analyst wants to quickly compare spectra at several different points on a machine just to get an idea of relative vibration at various machine measurement locations.

For all practical purposes during PMP surveys, 75% overlapping with only 2 averages need to be specified for F_{MAX} up to 3000 CPM (50 Hz), not 8 averages as used in the above example. Only 4 averages using about 50% overlap need to be specified for frequency ranges with an F_{MAX} between 3000 and 12,000 CPM (50 and 200 Hz); while 8 averages with 50% overlap for frequency ranges from 12,000 to 60,000 CPM (200 to 1000 Hz). For frequency ranges over 60,000 CPM (1000 Hz), 16 averages with no overlapping is recommended.

In the above example if the frequency were increased to 150,000 CPM (2500 Hz), overlapping would have had little value because the sampling time would have dropped from only 1.28 seconds (non-overlapped) down to 0.72 second (50% overlapped). But at 1500 CPM and 2 averages, overlapping does reduce the sampling time considerably. Thus it is a practical technique.

3.7 UNDERSTANDING A VIBRATION SPECTRUM

A spectrum is a graphic display of vibration in the frequency domain. Figure 16 is an example of a spectrum. Note that the X-axis is delineated in CPM with a maximum frequency (F_{MAX}) of 12,000 CPM and the Y-axis is delineated in in/sec of vibration with .050 in/sec peak/division (with 10 divisions, this gives a Full Scale Amplitude of .500 in/sec). The horizontal scale also has ten divisions with each frequency division being one-tenth of 12,000 CPM (or 1200 CPM/division). The addition of a cursor makes determining the frequency and amplitude of an individual peak easy. In Figure 16, the cursor is at the dominant peak. The cursor frequency is indicated as 1770 CPM (the approximate rotating speed of the machine) and its amplitude is .2474 in/sec. The dashed line across the bottom of the spectrum is an imaginary amplitude threshold level of 1%. That is, 1% of the full amplitude scale which is .500 in/sec which amounts to .005 in/sec. Below the spectrum is a listing of the amplitudes, the order (order = multiple of running speed), and the frequencies of all peaks that have an amplitude over .005 in/sec threshold.

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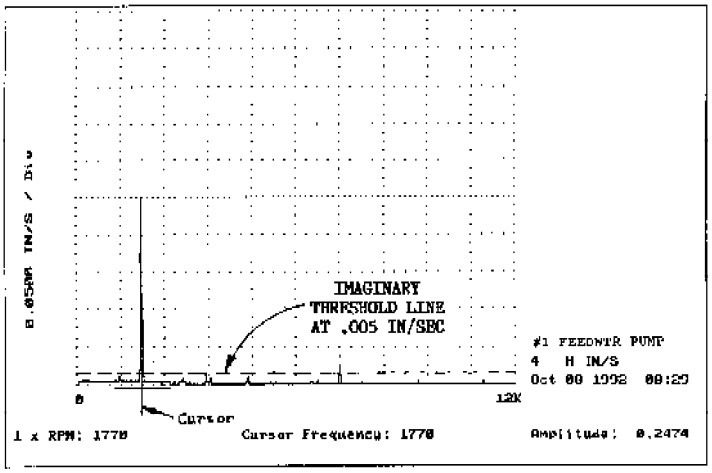
It is very important that both the amplitude and frequency values for all peaks in the spectrum be as precise as possible. Amplitude is important, of course, because it gives an idea of the severity of the problem. Frequency, on the other hand, is even more important since it is used to determine the source of the vibration. For instance, In the spectrum in Figure 16 the vibration with the highest peak (at 1770 CPM) equals the rotational speed of the machine (1X RPM). Therefore, the component's rotating speed at 1770 RPM is the source of the dominant vibration. By knowing that the problem source is at 1X RPM, all possible problem sources which cause vibration at 1X RPM can be itemized. At the same time, the possibility of other sources being the problem, such as rolling element bearing, blade pass, or most electrical problems, are eliminated.



1% x 0.500 = 0.0050 1 X RPM: 1770.00

Machine	Pos	Dir	Units	Date
#1 FEEDWTR PUMP	4	H	IN/S	Oct 08 1992 08:29

Frequency	1 x RPM	Amplitude
180	0.102	0.0054
240	0.136	0.0057
300	0.169	0.0070
1050	0.593	0.0062
1170	0.661	0.0088
1260	0.712	0.0053
1350	0.763	0.0053
1470	0.831	0.0053
1620	0.915	0.0051
1770	1.000	0.2474
2910	1.644	0.0078
3540	2.000	0.0144
3660	2.068	0.0055
4680	2.644	0.0108
7200	4.068	0.0234

T - Threshold adjust Q - Quit

FIGURE 16
SPECTRUM WITH CURSOR FREQUENCY AND IDENTIFICATION OF PEAKS ABOVE A USER-DEFINED THRESHOLD

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3.71 EFFECT OF THE NUMBER OF FFT LINES USED ON FREQUENCY ACCURACY

Probably the greatest requirement in determining potential machine problem source(s) is knowing the actual value of the frequency. The determination of an accurate vibration frequency can become clouded or enhanced, depending on how many lines of spectral resolution are used in collecting and displaying the data.

Most computer monitors are able to display 400 lines of spectral resolution (and most data collectors today are set up to capture 400-line spectra on PMP surveys). As discussed previously, the spectral display is divided up by 400 individual lines (or "bins") located adjacent to one another along the frequency axis. Not all bins contain information since the vibration at some frequency bin locations is zero. But the bins that do contain information display an amplitude peak(s) which is proportional to the amount of vibration at that specific bin location or frequency within it. The frequency range per bin (known as frequency resolution) is determined by Equation 5 below:

$$\text{FREQUENCY RESOLUTION} = \frac{\text{FREQ. SPAN}}{\text{\# FFT LINES}} \qquad \text{Eqn. 5}$$

Where:

FREQUENCY RESOLUTION = DISPLAY RESOLUTION PER LINE (CPM/LINE OR HZ/LINE)

FREQUENCY SPAN = Frequency Span from F_{MIN} to F_{MAX}; which is often 0 - F_{MAX} (CPM or Hz)

For example, in Figure 16, the F_{MAX} is 12K CPM (12,000 CPM). If it is divided by 400 lines of resolution, each bin has a 30 CPM range. The 59th bin actually contains vibration information in the range from 1755 to 1785 CPM. The 1770 CPM frequency is in this bin. But it is unknown whether there are one or more peaks within this bin. The vibration peak displayed is the resultant of all frequencies in this bin and the amplitudes of those frequencies within the bin all sum to .2474 in/sec. However, there may actually be only one frequency in this bin after all.

The higher the number of lines of resolution that are specified for a spectrum, the higher will be the precision of the frequency read by the analyzer and displayed. It is important to note that the precision (accuracy range) of any displayed frequency will be the frequency reading plus or minus one-half the resolution. Thus, if a peak at 1800 CPM was displayed in a 12,000 CPM spectrum with a 30 CPM frequency resolution, the accuracy range would be 1800 CPM (±15 CPM), or from 1785 to 1815 CPM. On the other hand, if a 120,000 CPM spectrum using 400 lines was taken at the same location and showed the same 1800 CPM peak, the accuracy would be ± half the resolution (300 CPM), or from 1650 to 1950 CPM. Figures 17 thru 20 are spectra with 100, 200, 400, and 3200 lines of resolution. Note that the peaks in the 100-line spectrum of Figure 17 have a very "blocked" appearance with that dominant peak shown at "1800 CPM". With 100 lines of resolution, each bin has a frequency resolution (sometimes called "Δf") of 120 CPM per line of resolution. Therefore, the frequency value from a 100 line spectrum is considerably inaccurate. Figure 18 has 200 lines of resolution. As a result, its resolution of 60 CPM/line is slightly better than that of Figure 17. Here, the dominant frequency is observed to be at "1740 CPM". Figure 19 is a 400-line spectrum with 30 CPM/line of resolution. The peak is identified as "1770 CPM", which is more accurate than the values in Figures 17 and 18.

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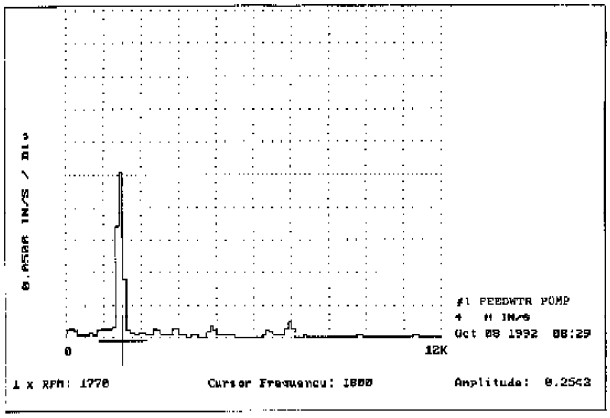


001% x 0.500 = 0.0050 1 x RPM: 1770.00

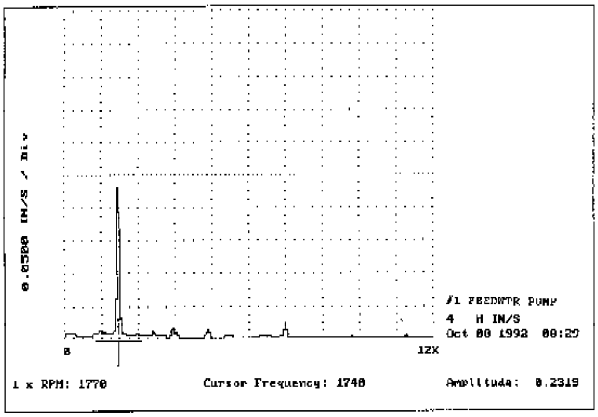
Machine	Pos	Dir	Units	Date
#1 FEEDWTR PUMP	4	H	IN/S	Oct 08 1992 08:29

Frequency	1 x RPM	Amplitude
240	0.136	0.0120
840	0.475	0.0053
1200	0.678	0.0132
1800	1.017	0.2542
2400	1.356	0.0061
2880	1.627	0.0115
3600	2.034	0.0134
4680	2.644	0.0154
5480	3.661	0.0081
7200	4.068	0.0227

FREQ. RESOLUTION = $\frac{12{,}000 \text{ CPM}}{100 \text{ LINES}}$ = 120 CPM/LINE

T - Threshold adjust Q - Quit

FIGURE 17
12,000 CPM SPECTRUM WITH 100 LINES OF RESOLUTION



1% x 0.500 = 0.0050 1 x RPM: 1770.00

Machine	Pos	Dir	Units	Date
#1 FEEDWTR PUMP	4	H	IN/S	Oct 08 1992 08:29

Frequency	1 x RPM	Amplitude
120	0.068	0.0076
300	0.169	0.0072
1140	0.644	0.0111
1320	0.746	0.0087
1500	0.847	0.0075
1740	0.983	0.2319
2400	1.356	0.0051
2880	1.627	0.0083
3540	2.000	0.0160
4680	2.644	0.0116
7200	4.068	0.0237

FREQ. RESOLUTION = $\frac{12{,}000 \text{ CPM}}{200 \text{ LINES}}$ = 60 CPM/LINE

T - Threshold adjust Q - Quit

FIGURE 18
12,000 CPM SPECTRUM WITH 200 LINES OF RESOLUTION

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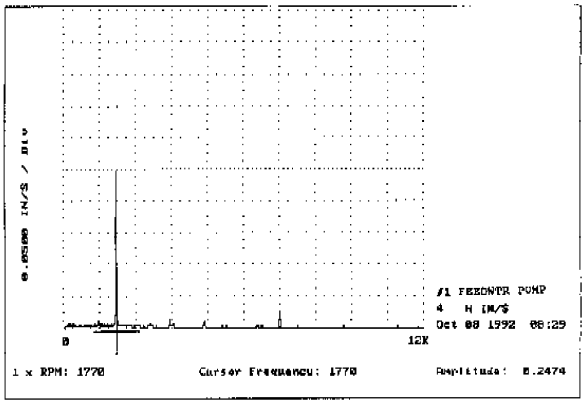


1% x 0.500 = 0.0050 1 x RPM: 1770.00

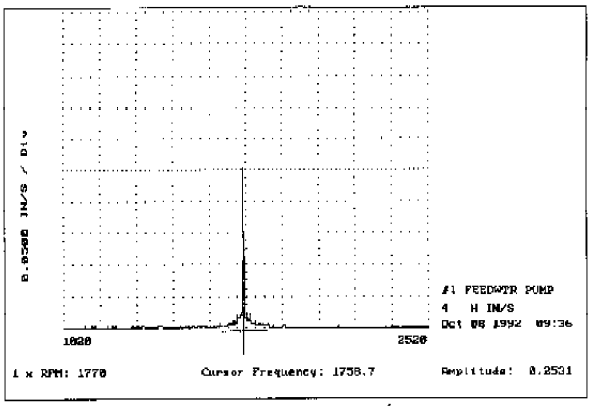
Machine	Pos	Dir	Units	Date
#1 FEEDWTR PUMP	4	H	IN/S	Oct 08 1992 08:29

Frequency	1 x RPM	Amplitude
180	0.102	0.0054
240	0.136	0.0057
300	0.169	0.0070
1050	0.593	0.0062
1170	0.661	0.0088
1260	0.712	0.0053
1350	0.763	0.0053
1470	0.831	0.0053
1620	0.915	0.0051
1770	1.000	0.2474
2910	1.644	0.0078
3540	2.000	0.0144
3660	2.068	0.0055
4680	2.644	0.0108
7200	4.068	0.0234

FREQ. RESOLUTION = $\frac{12,000 \text{ CPM}}{400 \text{ LINES}}$ = 30 CPM/LINE

T - Threshold adjust Q - Quit

FIGURE 19
12,000 CPM SPECTRUM WITH 400 LINES OF RESOLUTION



006% x 0.500 = 0.0300 1 x RPM: 1770.00

Machine	Pos	Dir	Units	Date
#1 FEEDWTR PUMP	4	H	IN/S	Oct 08 1992 09:36

Frequency	1 x RPM	Amplitude
1758.7	0.994	0.2531

FREQ. RESOLUTION = $\frac{12,000 \text{ CPM}}{3200 \text{ LINES}}$ = 3.75 CPM/LINE

T - Threshold adjust Q - Quit

FIGURE 20
ZOOM SPECTRA TAKEN FROM A 12,000 CPM FREQUENCY SPAN SPECTRUM HAVING 3200 LINES OF RESOLUTION

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Further accuracy can be achieved by using higher numbers of lines of resolution. Figure 20 is a portion of a 3200-line spectrum. The resolution is 3.75 CPM/line of resolution for the 12,000 CPM frequency span. The dominant vibration is now shown as "1758.7 CPM". The accuracy (precision limits) of this dominant peak would be ±1.875 CPM (or, again, one-half the frequency resolution, Δf). Thus, the true frequency would actually be between 1756.8 and 1760.6 CPM (29.281 - 29.344 Hz).

So why shouldn't a 3200-line spectra be taken all the time and avoid the hassle of inaccurate data? First of all, a 3200-line spectrum takes 8 times longer to collect than does a 400-line spectrum (as seen by Equation 6 below) which really slows down the data collection process. Secondly, it requires 8 times the storage for a 3200 lines spectrum than for a 400-line spectrum. Judgement has to be exercised since the data collector's spectral storage capacity will be significantly reduced if such high resolution spectra are collected at all points. This will be discussed in more detail in a later section.

When would it be advantageous to take 3200-line spectra? To even detect certain electrical problems (such as cracked/broken rotor bars or end rings) will require high resolution spectra in order to even detect these problems as will be discussed in Chapter 6.12. It is also extremely useful when vibration frequencies from different machines are very close to one other. In other instances, some bearing frequencies and harmonics are commonly close to exact multiples of the operating speed. In another case, a pump may have an impeller with 6 vanes and rolling element bearings which have a defect frequency of 6.03X RPM. In this case, it may require a high resolution spectrum to separate the bearing frequency (6.03X RPM) from the vane passing frequency (6X RPM). Here, a high degree of accuracy is needed. However, for general data collection, 400 lines of resolution produces adequate accuracy.

$$DATA\ SAMPLING\ TIME = \frac{60(\#\ FFT\ LINES)(\#\ AVG.)}{F_{MAX}} \qquad \text{Eqn. 6}$$

Equation (6) is a modification of Equation (4).

Where:

DATA SAMPLING TIME	=	Time to calculate FFT from Time Waveform (sec); Assuming no Overlap Processing.
# FFT LINES	=	Number of FFT Lines or Bins in Spectrum
# AVG.	=	Number of Averages
F_{MAX}	=	Frequency Span (CPM)

3.72 EFFECT OF THE FREQUENCY SPAN USED ON FREQUENCY ACCURACY

Another factor affecting resolution is the frequency range (F_{MAX}) setting. The larger the frequency range, the less accurate the frequency readings will be and the wider each line of resolution (bin) will be. Figure 21 is a 400-line spectrum with a 24,000 CPM F_{MAX}. The resolution of this spectrum is 24,000/400 lines of resolution = 60 CPM/line of resolution. In comparison with the 12,000 CPM F_{MAX}, 400-line spectrum of Figure 19, its frequency definition is only one-half as accurate. Figure 22 is a 120,000 CPM F_{MAX} spectrum with 400 lines of resolution. Here, the resolution is 300 CPM/line of resolution which means that the accuracy of any displayed frequency will be ±150 CPM (or plus or minus one-half the resolution). In this case, its frequency definition is only 1/10 of the precision in Figure 19. In some cases, this amount of accuracy may be all that is needed.

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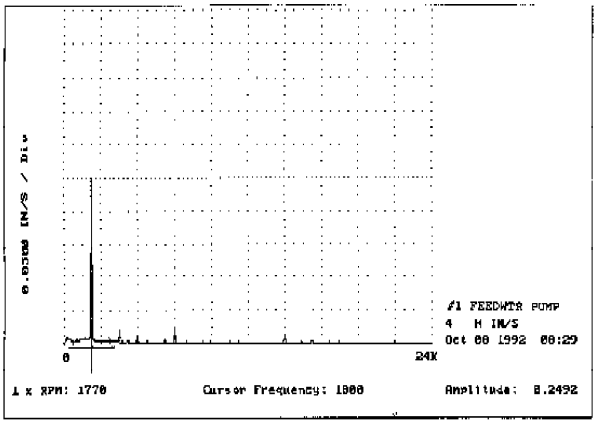


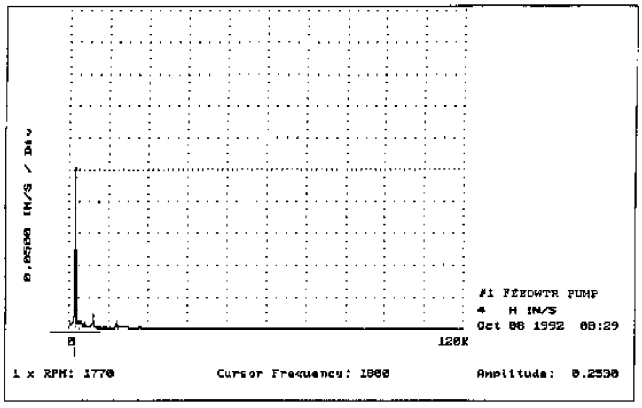
1% x 0.500 = 0.0050 1 x RPM: 1770.00

Machine	Pos	Dir	Units	Date
#1 FEEDWTR PUMP	4	H	IN/S	Oct 08 1992 08:29

Frequency	1 x RPM	Amplitude
180	0.102	0.0095
780	0.441	0.0050
1020	0.576	0.0066
1140	0.644	0.0076
1380	0.780	0.0097
1500	0.847	0.0078
1800	1.017	0.2492
2940	1.661	0.0083
3600	2.034	0.0178
4740	2.678	0.0105
7200	4.068	0.0253
14400	8.136	0.0111

FREQ. RESOLUTION = $\frac{24,000 \text{ CPM}}{400 \text{ LINES}}$ = 60 CPM/LINE

FIGURE 21
24,000 CPM SPECTRUM WITH 400 LINES OF RESOLUTION



1% x 0.500 = 0.0050 1 x RPM: 1770.00

Machine	Pos	Dir	Units	Date
#1 FEEDWTR PUMP	4	H	IN/S	Oct 08 1992 08:29

Frequency	1 x RPM	Amplitude
300	0.17	0.0095
1800	1.02	0.2530
3000	1.69	0.0117
3600	2.03	0.0137
4800	2.71	0.0091
7200	4.07	0.0230
14400	8.14	0.0111

FREQ. RESOLUTION = $\frac{120,000 \text{ CPM}}{400 \text{ LINES}}$ = 300 CPM/LINE

FIGURE 22
120,000 CPM SPECTRUM WITH 400 LINES OF RESOLUTION

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Trade-offs must be made to retain accuracy. The analyst needs to specify an F_{MAX} large enough to include all the vibration data needed, while not (at the same time) unduly sacrificing the accuracy in the process. Technical Associates has a proven method for choosing F_{MAX} values for data collectors. This method will be discussed in detail in a later chapter.

3.73 IMPROVING THE FREQUENCY RESOLUTION WITH "ZOOM" - BAND SELECTABLE FOURIER ANALYSIS

Recall that a Fast Fourier Transform is equivalent to a set of filters, starting at zero Hertz, equally spaced up to some maximum frequency. Therefore, the frequency resolution is limited to the maximum frequency divided by the number of "filters".

In addition to the examples given in Section 3.71, suppose we needed to measure a small signal which is very close in frequency to a large one. For example, the condition of an electric motor can be assessed by acquiring motor current spectra using a current transformer (CT) as the transducer. When evaluating motor current spectra for rotor problems, one will have to compare the amplitude at electrical line frequency (F_L) with that at what is known as the first pole pass frequency (F_P) to its left (where pole pass frequency = slip frequency X # poles as will be discussed in Chapter 6.12). In this country, the line frequency (F_L) is normally 60 Hz (3600 CPM) and will dominate motor current spectra, thereby requiring the analyst to use "log amplitude" to even see the pole pass frequency sidebands surrounding it. And, typically these pole pass frequencies will only range from about 30 CPM (0.5 Hz) to 180 CPM (3.0 Hz). Therefore, precise frequency resolution is required for this measurement. One obvious advantage in decreasing the frequency resolution is the ability to distinguish closely spaced peaks in the spectrum.

Two ways of improving the frequency resolution were discussed in the last section. One way to improve frequency resolution is to increase the number of FFT lines in the spectrum and then display only those lines of interest. Another way is "frequency zoom". Frequency zoom is the ability to concentrate all 400 lines of resolution in the frequency band of interest. Figure 23 is a display of the example above. Figure A is the measurement without using zoom. It is a partial display (40 lines of a 400 line spectrum that spans from 0 to 200 Hz or 0 - 12,000 CPM). Figure B is the same measurement using zoom. With zoom, all 400 lines of the spectrum are concentrated between 50 and 70 Hz (3000-4200 CPM given a 3.0 CPM resolution). In Figure A, the sidebands are barely above the noise floor - if they were much lower they would not be measurable. The noise floor in Figure 23B is reduced, making the sidebands much more visible.

There is a difference between "zoom" and "true zoom". Some analyzers can compute spectra with up to 12,800 lines of interest. If only a portion of the 12,800 lines are displayed, the result is the same as zoom measurement. Although some vendors may call this zoom, it is actually "display expansion". It is not "true zoom". True zoom is implemented by selecting the frequency span of interest before performing the FFT, not afterwards. Thus, a 400 line FFT would be adequate for most "true zoom" measurements because all 400 lines of resolution and concentrated between the frequencies of interest (F_{MIN} to F_{MAX}).

But the frequency resolution would not be improved by selecting a so-called "zoom" from a spectrum stored with only 400 FFT lines. By selecting a scaling factor of 4 on the analyzer only expands the frequencies by that factor proportionally. As shown in Figure 24, F_{MAX} was made to equal 15,000 CPM rather than the original 60,000 CPM. The frequency expanded spectrum allows for a better look at the shape of each frequency peak, but it does not improve the frequency resolution.

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Figure A - Spectrum of motor current, measured without zoom

DISPLAY IS ONLY 40 LINES OF A 400-LINE FFT

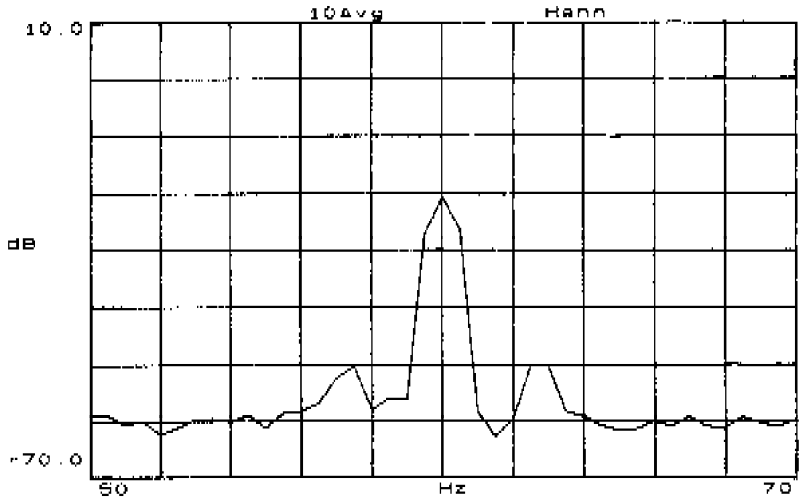

Figure B - Spectrum of motor current, measured with zoom

DISPLAY IS 400 LINES CONCENTRATED (AROUND 60 Hz) BETWEEN 50 TO 70 Hz

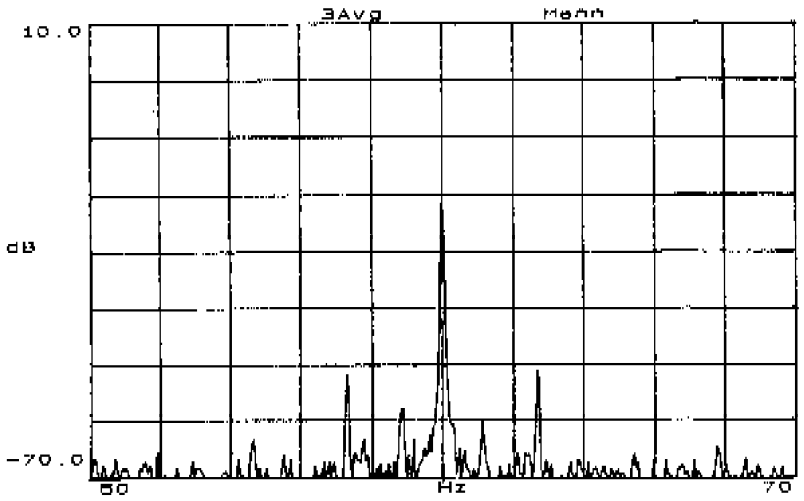

FIGURE 23
TWO EXAMPLES OF MOTOR CURRENT SPECTRA, WITH AND WITHOUT "ZOOM" (Ref. 5)

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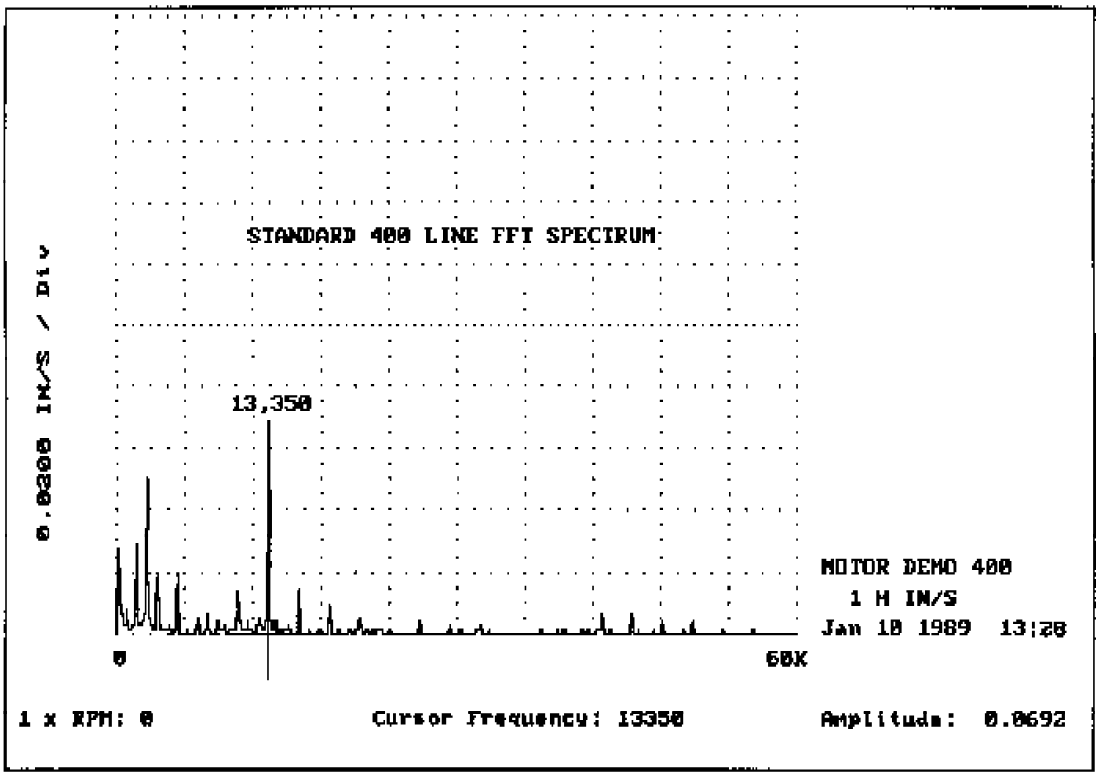

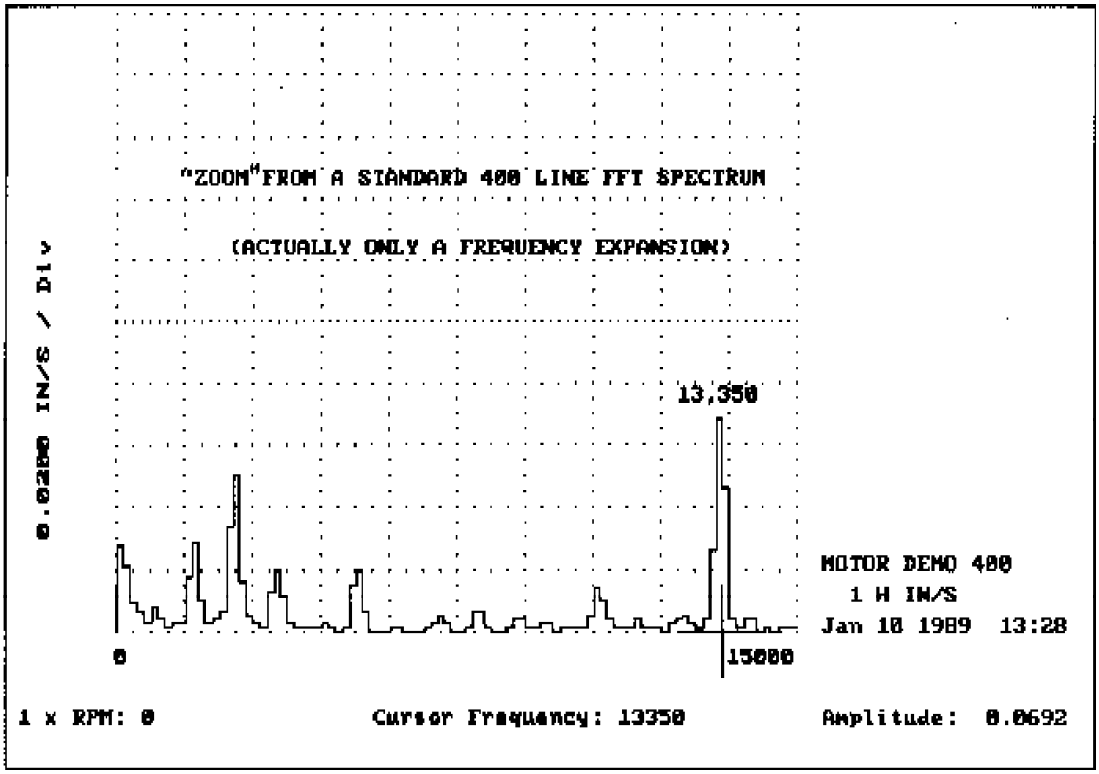

FIGURE 24
EXAMPLE OF A SO-CALLED "ZOOM" SPECTRUM

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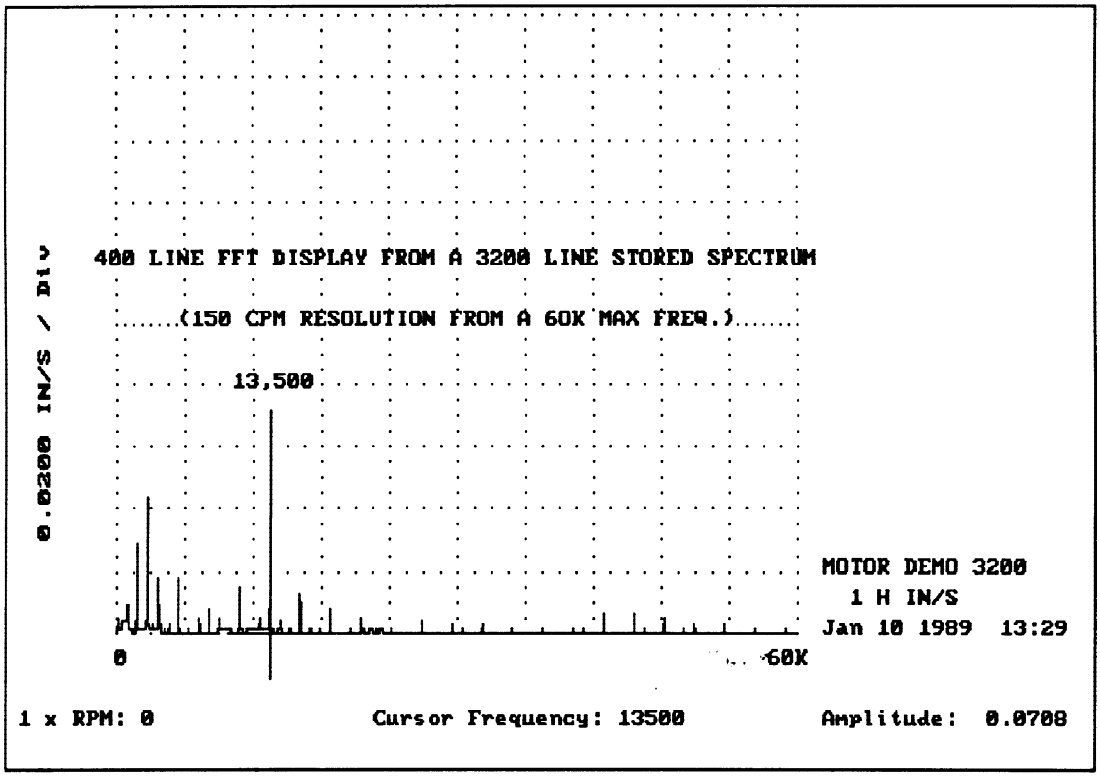

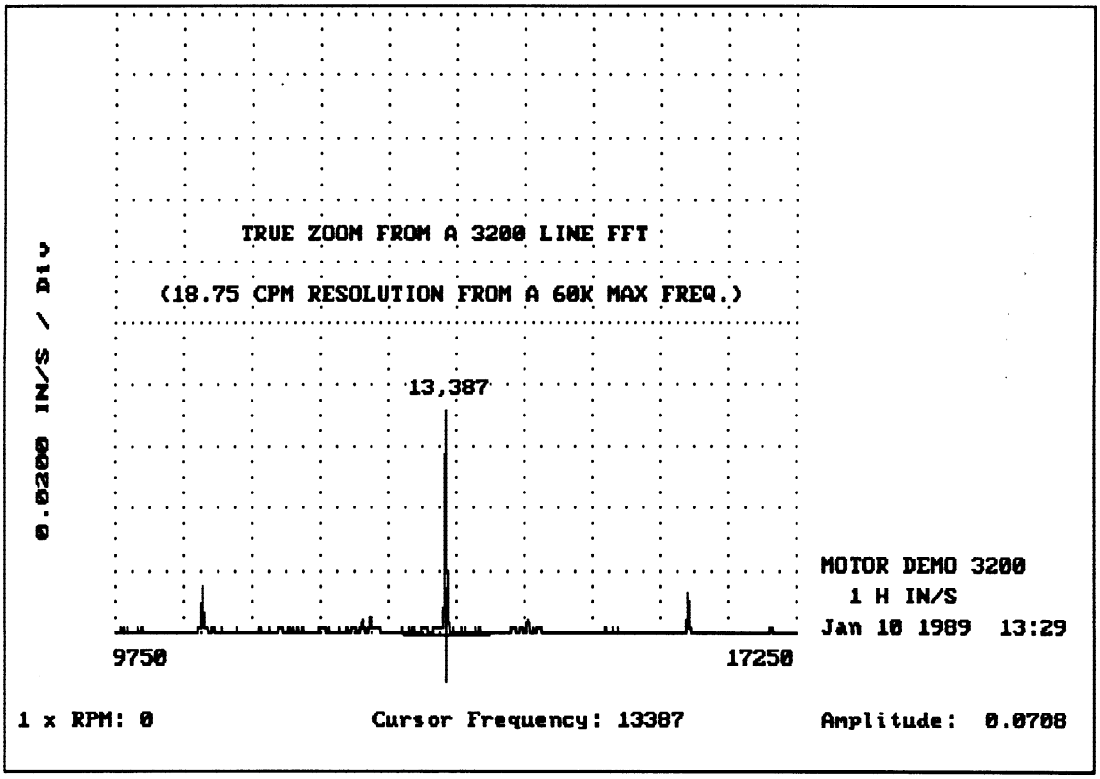

FIGURE 25
EXAMPLE OF A "TRUE ZOOM" SPECTRUM (Ref. 5)

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As was mentioned previously, one way to improve resolution is to increase the number of lines in the FFT and then display them. For instance, capturing and storing a 3200-line FFT does improve frequency resolution by a factor of 8X (3200/400) as shown in Figure 25. However, a 3200-line FFT also requires 8X the measurement time in the field and requires 8X the storage space of a 400 line FFT back in the computer.

Also mentioned previously, was selecting the frequency band of interest before performing the FFT. This is practical if the frequency span of interest is known before starting the measurement, provided the analyzer is capable of performing this band selectable Fourier analysis. The advantages are apparent. First, "true zoom" can reduce the measurement time considerably and reduce data storage because a 400 line FFT only takes 1/8 the memory, compared to 3200 lines. However, in the real-world, the frequency span which needs to be zoomed is rarely known, unless it was derived from previous data. In this case, it is better to use display expansion, a pseudo "true zoom", because the improved frequency resolution will accurately identify specific peaks of interest as shown in Figure 25. Here, the 13,500 CPM peak seen in the upper 400-line display, is more accurately identified as a 13,387 CPM peak in the "zoom" of the lower display using 3200 lines. Also, the sidebands can be readily identified.

The point to remember is merely performing a display expansion on a "zoom' of a 400-line FFT will not approve the frequency resolution. Only by increasing the lines of resolution and/or decreasing the F_{MAX}, will the "zoom", "display expansion" or "true zoom" actually improve the frequency resolution so that spectral peaks can be seen and accurately identified.

3.74 IMPROVING THE PRECISION OF THE SPECTRUM BY FREQUENCY & AMPLITUDE INTERPOLATION

Only the latest vibration instruments and software are capable of displaying as much as 12,800 line spectra. A technique which can be employed by an analyst to enhance the frequency and amplitude values from common 400 line spectral data will be discussed in this section. A set of equations will be provided to improve frequency and amplitude readings by as much as 100 times, given any frequency span and the number of FFT lines of resolution. Of course, for a smaller frequency span with a greater number of FFT lines, the calculated values will be closer to the actual value.

$$f_I = f_p + \Delta f_c \quad (if\ f_s > f_p) \qquad \text{Eqn. 7A}$$

$$f_I = f_p - \Delta f_c \quad (if\ f_s < f_p) \qquad \text{Eqn. 7B}$$

3.741 Equations for Closest Frequency Interpolation (f_I)

where:

- f_I = Interpolated Frequency calculated to be closest to the True (actual) Frequency, dependent on Frequency Resolution, Δf (see Figure 26 (A-2))
- f_p = Displayed Frequency on Spectrum (Closest Frequency Peak to f_I, dependent on Frequency Resolution used, Δf)
- Δf_c = Frequency Correction, or distance from f_p to f_I (CPM or Hz)
- f_s = Frequency (either above or below f_p) in the next bin (f_s) (see Figure 26 (A-1))

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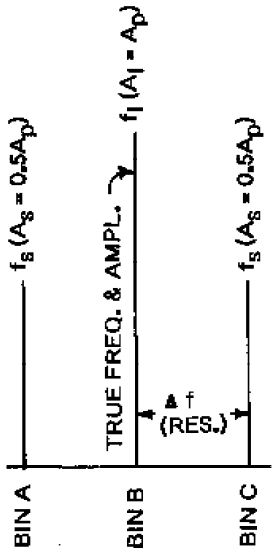

Fig. A-1

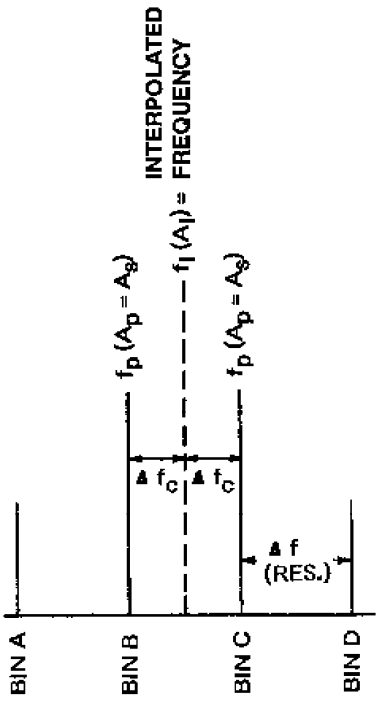

Fig. A-2

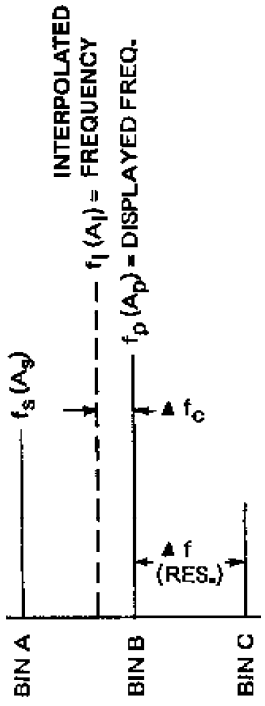

Fig. A-3

FIGURE 26
IMPROVEMENT OF DISPLAYED AMPLITUDE & FREQUENCY READINGS BY BIN DISTRIBUTION CALCULATIONS (SEE EQUATIONS 7A THRU 9)

$$\Delta f_c = \frac{2A_s - A_p}{A_s + A_p}\Delta f \qquad \text{Eqn. 8}$$

The Frequency Correction (Δf_c) can be calculated as follows:

where:

A_s = Amplitude of the Frequency in Bin f_s (see Figure 26 (A3))
A_p = Amplitude of the Frequency in Bin f_p (see Figure 26 (A3))
Δf = Frequency Resolution = FREQ. SPAN/#FFT LINES (Hz or CPM/Line)

Therefore, by using Equations 7 and 8, the precision of the displayed frequencies can be improved very much by examining closely the amplitudes of the bins to the right and left of each frequency of interest one at a time, choosing the higher amplitude of the two bins adjacent to it. Examples that follow will demonstrate the use of this technique, such as in Figures 27 and 28.

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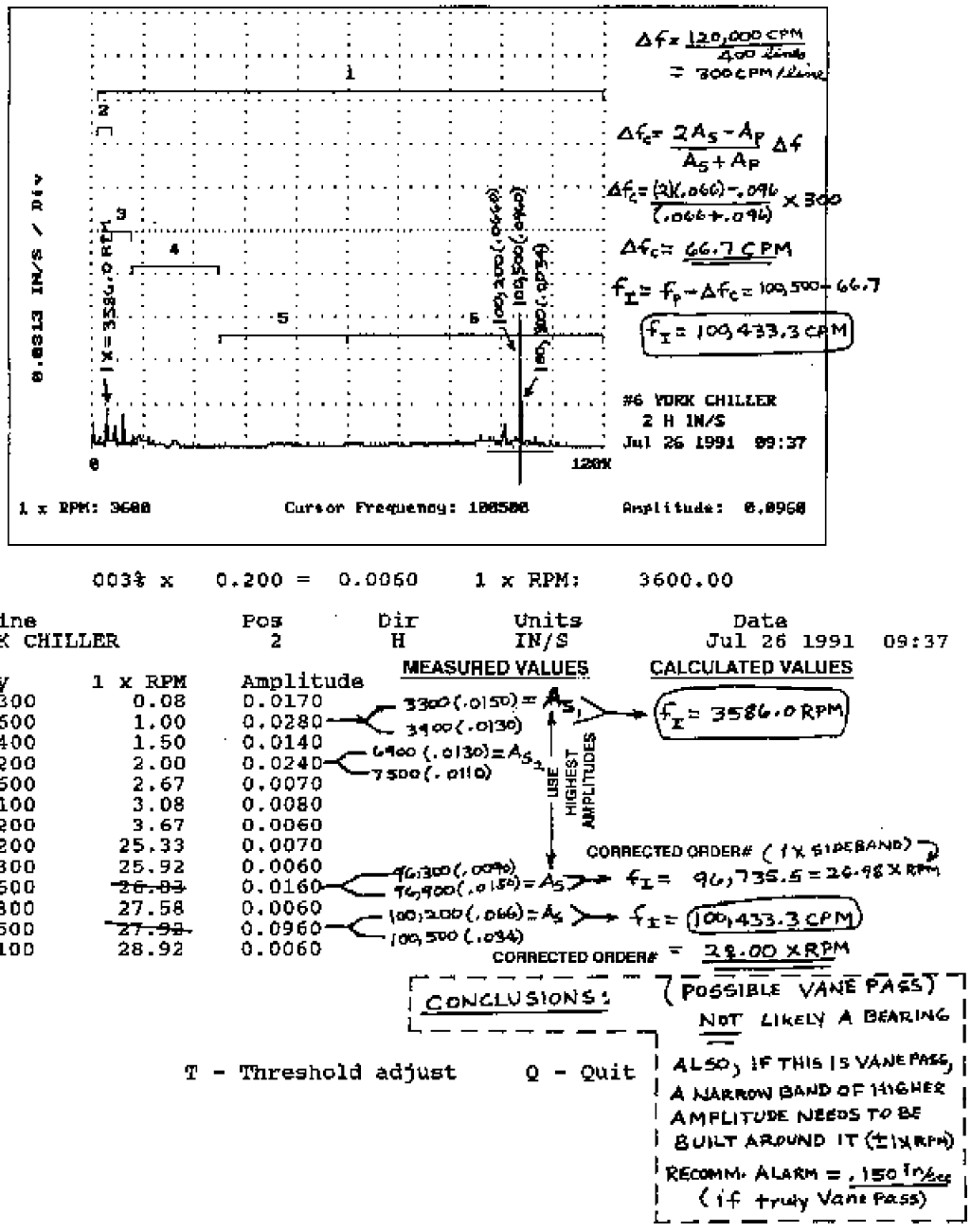

FIGURE 27
EXAMPLE OF USING EQUATIONS FOR FREQUENCY INTERPOLATION FROM MEASURED FREQUENCY AND AMPLITUDE VALUES

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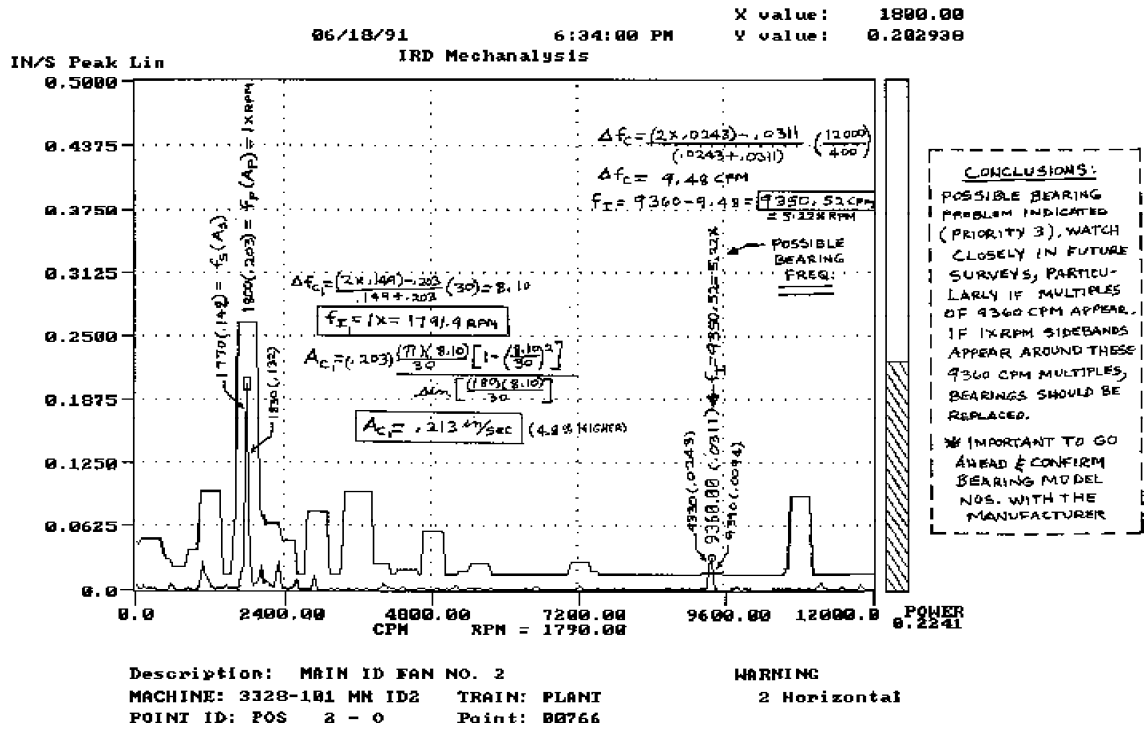

FIGURE 28
EXAMPLE OF USING EQUATION FOR AMPLITUDE INTERPOLATION FROM MEASURED FREQUENCY AND AMPLITUDE VALUES

3.742 Equation for Closest Amplitude Interpolation (A_I)

$$A_I = A_p \frac{\frac{\pi \Delta f_c}{\Delta f}\left[1-\left(\frac{\Delta f_c}{\Delta f}\right)^2\right]}{\sin\left[\frac{180(\Delta f_c)}{\Delta f}\right]} \qquad \text{Eqn. 9}$$

Similarly, amplitude readings can be improved by examining the same data shown in Figure 27 as follows:

where:

A_I = Interpolated Amplitude calculated to be closest to the True Amplitude of the True Frequency, dependent on the Frequency Resolution used.

A_p = Amplitude of Closest Frequency Bin (f_p) to the True Frequency, (where units of amplitude can be g, in/sec, mils, dB, etc.)

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3.75 IMPROVING THE AMPLITUDE ACCURACY BY CHECKING THE BANDWIDTH

The previous technique will have limitations any time that two frequencies exist within one bandwidth. For example, in the case where a 2-pole motor has a running speed of 3590 RPM: if a frequency span of 60,000 CPM and 400 lines of resolution is employed, the 48th line (or bin) will contain both the 2X RPM (7180 CPM) and the 2X electrical line frequencies (7200 CPM). This will adversely affect the interpolated calculation of the amplitude of the peak and the value of frequencies (in fact, the frequency interpolation technique will not work if two or more frequencies are either within one bin or in two bins adjacent to one another).

Recall from Sections 3.41 and 3.42 that the mathematics behind the Fourier Transform assumes that the input data is periodic. (That is, an exact integer number of periods of input data will be supplied). Rarely is this the case. Instead, there is generally non-periodic input data that is supplied to the transform. Without windowing, the FFT will find the discontinuity, between the end of one sample (data buffer) and the beginning of the next, and calculate the extra frequency components that appear as sidebands about the frequency of interest. Recall that the calculation of these extra frequencies is known as “spectral leakage”.

These Sections also showed that the spectral leakage can be reduced by multiplying the time sample (the frame of time domain data going to the FFT algorithm) by a window, shaped to smooth the transition between the end of one time sample and the beginning of the next. Recall from Section 3.43 that the most commonly used window is the Hanning Window (it is also known as the cosine squared window; it has zero magnitude and slope at both ends, and a magnitude of 1 in the middle). That is, the FFT is forced to concentrate on the center portion of the sampled data. But even with the Hanning Window, some leakage will be present when an FFT is performed on a non-integer number of cycles of a signal. What this implies is that the frequency component of a signal does not exactly correspond to one of the lines (bins) in the frequency spectrum.

For example, the Hanning window can spread a single sine wave across three spectral lines, whereas a single sine wave would appear as a single spectral line if no window is used. Even though windowing reduces spectral leakage, there is still some spreading and reduced amplitude when any frequency does not coincide exactly with the center of an FFT line (or bin).

This phenomenon presents a problem. If a single sine wave can be spread across three (or four) lines, how can one realistically determine what the narrowest bandwidth can be so as to be able to separately display two frequencies (closely spaced to one another), each showing its peak amplitude?

The following calculation is a further refinement of what has been presented so far. At this point, we are familiar with the definition of frequency resolution as being the frequency span (used) divided by the number of FFT lines. What needs to be done next is to modify the frequency resolution or “bandwidth” by introducing a window factor as follows:

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$$\text{BANDWIDTH} = \frac{(\text{FREQUENCY SPAN})(\text{WINDOW NOISE FACTOR})}{\#\ \text{FFT LINES}} \qquad \text{Eqn. 10}$$

Where:

BANDWIDTH = The spacing between frequencies at which a bandpass filter attenuates the signal by 3 dB (down to 70.7% of its flat response). Two individual frequencies must have at least two bandwidths separation for them both to be displayed on an FFT spectrum with each showing its actual "peak" amplitude.

WINDOW NOISE FACTOR = Effective Noise Bandwidth
= 1.5 for Hanning Window
= 1.0 for Uniform Window (also called Rectangular Window)
= 3.6 - 4.2 for Flat Top Window (depending on analyzer vendor)

If no window (Uniform window) is used, one would use 1.0 for the window factor, whereas if a Hanning window is used, the window factor is 1.5. If other windows are used in an analysis, be aware that the results from Equation 8 (calculation of Δf_c frequency correction) and Equation 9 (calculation of Interpolated Amplitude A_I) will be altered. To simplify this discussion, only the window factor for the Hanning window will be highlighted.

It would appear that by multiplying the resolution by 1.5, the bandwidth would be 1.5 times larger than it would be if the frequency resolution was left as is. Mathematically, that is true. However, do not miss the purpose of this exercise. It is done to determine the narrowest bandwidth that can be used to separately display two frequencies closely spaced to each other. It is meant to compensate for the fact that the signal was spread across three spectral lines, instead of in one line, since a Hanning window was used. Especially when setting up a database, what we are trying to determine is, if we are aware of the presence of closely-spaced frequencies, what would the maximum bandwidth have to be in order to display two distinct frequencies? This would particularly apply in the analysis of AC induction motors. The presence of closely-spaced pole-pass frequency sidebands around 1X RPM or around the 2X, 3X, 4X and 5X RPM harmonic peaks is a strong indication of broken or cracked rotor bars and/or high resistance joints at the shorting ring. Also, in the case of a two-pole motor, 2X RPM will be extremely close to the 7200 CPM (2X F_L) peak. Or, in the case of a four-pole motor, the 4X RPM peak will be extremely close to the 7200 CPM peak.

For most of the frequency ranges chosen, an analyzer will not display two individual frequencies; in these examples, it may display only one frequency in the above cases. In order to see two separate closely-spaced frequencies, it will be necessary either to increase the number of FFT lines, lower the frequency span, or generate a "zoom" spectrum.

This technique is illustrated in Figure 29. Figure 29A is a spectral plot from a 2-pole AC motor driving a centrifugal air compressor. The measurement point is at the inboard bearing of the motor in the horizontal direction. The F_{MAX} was set at 30,000 CPM using 400 lines of resolution in the analyzer. Thus, the resolution was 75 CPM/line. Multiplying this value by 1.5 gives us a bandwidth of 112.5 CPM. Notice that all the peaks are certainly more than 112.5 CPM apart. In this case, it appears that these peaks are running speed harmonics of 3600 CPM (approximately 1X RPM). In fact, this particular motor was originally diagnosed by this client as having a mechanical looseness problem.

However, a further analysis was done. Using a real-time analyzer, the frequency span was now set to 6000 CPM (100 Hz), but the cursor was set on the 1X RPM peak and a zoom was initiated. Figure 29B is a spectral plot of this zoom, which is 600 CPM wide over 400 lines of resolution.

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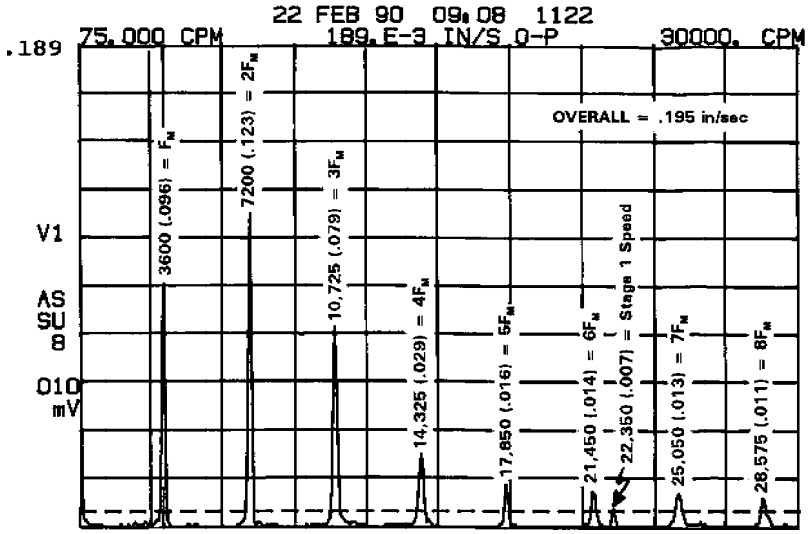
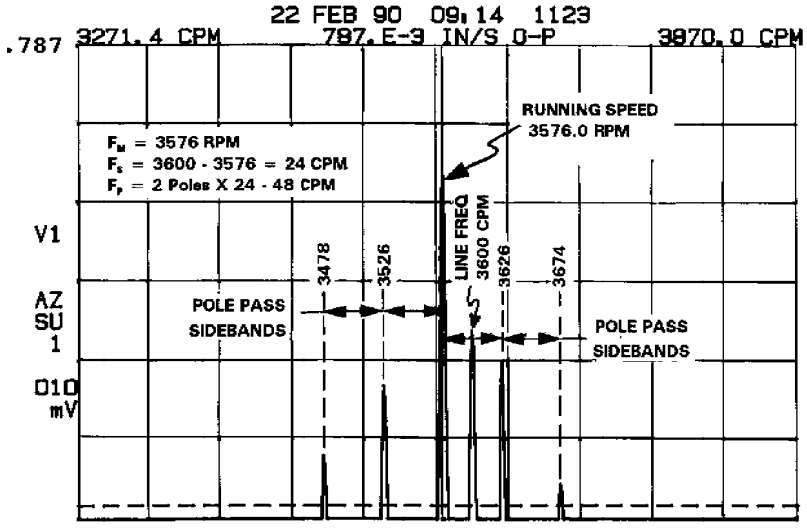
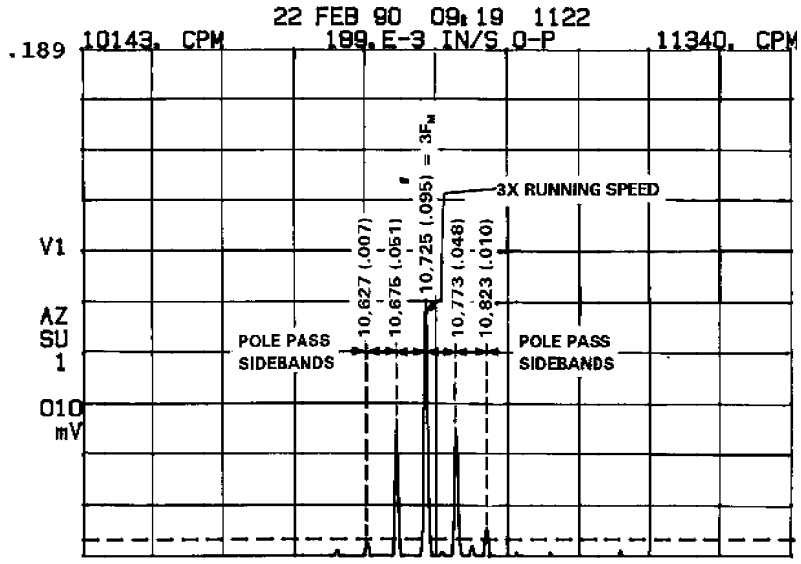


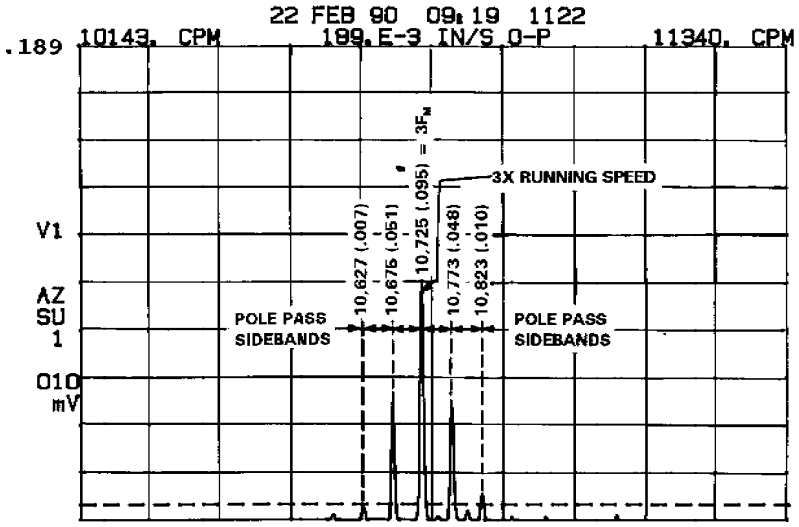
29A- 30,000 CPM SPECTRUM ON A CENTAC MOTOR (POS. 2H)



29B- ZOOM SPECTRUM AROUND 1X RPM SHOWING MULTIPLE POLE PASS SIDEBANDS (BROKEN OR CRACKED ROTOR BARS/ SHORTING RING PROBLEMS INDICATED)



29C- ZOOM SPECTRUM AROUND 2X RPM
(POLE PASS SIDEBANDS ALSO HERE)



29D- ZOOM SPECTRUM AROUND 3X RPM
(NOTE POLE PASS SIDEBANDS HERE ALSO)

FIGURE 29
WIDEBAND AND ZOOM SPECTRA FOR A MOTOR HAVING CRACKED OR BROKEN ROTOR BARS OR SHORTING RING PROBLEMS

The resolution is 1.5 CPM/line and the bandwidth is 2.25 CPM. In this case, pole-pass sidebands can clearly be seen around the 1X RPM peak, (two "families" of them in fact). In addition, the line frequency (F_L = 3600 CPM) was also present. These peaks certainly were not evident in Figure 29A.

Similarly, Figure 29C is a spectral plot of a zoom around the 2X RPM peak. The frequency span was now set to 12,000 CPM (200 Hz), but the zoom displays 1200 CPM (10% of 12,000 CPM) over 400 lines. Therefore, the resolution is 3 CPM/line and the bandwidth is 4.5 CPM. Once again, the pole-pass frequencies can be clearly identified around 2X RPM. These peaks were also not evident in Figure 29A.

Likewise, Figure 29D is a spectral plot of a zoom around the 3X RPM peak. In this case, the resolution is also 3 CPM/line and the bandwidth is 4.5 CPM. Two "families" of pole-pass frequencies can be seen with a third just starting up. Notice that the amplitudes varied from .007 in/sec to .051 in/sec. Does this motor have an electrical problem or not?

It is generally accepted that if pole pass frequencies (F_P) are present around 3 or 4 running speed harmonics with amplitudes above .012 - .015 in/sec, the motor likely has serious problems. But how will we be able to see these pole pass sidebands if we don't consider the bandwidth first?

There may be a number of practical reasons why bandwidth needs to be given consideration. The most outstanding purpose behind calculating the bandwidth is to assure that there will be enough frequency separation to clearly define two or more spectral peaks closely-spaced as was seen with pole-passing frequencies. Another reason, but lesser in importance, is that the amplitude of a sideband in relation to the fundamental frequency is often used as a "gauge" to determine the seriousness of a defect or problem. Thus, it is important that there is enough resolution and bandwidth to be able to display the sideband frequencies so that their amplitudes can be measured. Coupled with zoom analysis, this technique provides the analyst with a means of displaying all necessary spectral peaks.

3.76 EFFECT OF DYNAMIC RANGE ON FREQUENCY AND AMPLITUDE DISPLAY

Apart from specifying the optimum F_{MAX}, number of FFT lines, and checking the bandwidth, there is another problem which could be responsible for not being able to see sidebands and other small peaks in the presence of larger peaks. The problem could be that the instrument used does not have enough "dynamic range".

Dynamic range is defined as "the amplitude ratio (in dB) between the maximum input level and the self-noise (or noise floor) of an instrument or system before distortion." The noise floor is that area along the spectrum where tiny spectral peaks can be seen. If the amplitudes of the peaks of interest are high enough to stand out like "trees in the grass", they are often easy to measure and evaluate. The problem becomes more complex when the spectral peaks are barely distinguishable from the noise floor; it then becomes difficult to see discrete frequencies. This is a primary concern, especially when performing low frequency, low amplitude measurements.

The vibration signal is analogous to a ship on an ocean, where sea level is equivalent to the noise floor of the measurement. The higher the ship rides in the water, the more information about it will be optically available and the easier it is to detect on the horizon - submerged ships go undetected. If a picture is taken of the ship, no amount of photographic enhancement will display the part of the ship which is below the surface or of any portion of submerged ships. The point is, noise can obscure spectral data, alter amplitude information, and render measurements useless. It is important to use an instrument that is equipped to maximize the signal-to-noise ratio of the vibration measurement. Therefore, the dynamic range on an instrument vendor's specification sheet needs to be given more consideration. In order to do this, one has to become familiar with terms, such as "dB", "bit", and "dynamic range".

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For instance, one might read that a certain vendor sells an instrument that is an “8-bit data collector with a dynamic range of 48 dB” or another vendor will sell “a 12-bit or 14-bit instrument with a dynamic range of 72 dB and 84 dB", respectively. What does this mean? Does it make any difference?

Recall from Section 3.1 that microprocessors work in powers of 2, called binary numbers. Thus, 1024 intervals or equally-spaced samples equal 2^{10} intervals. If an analog-to-digital instrument was designed to do this, it would be referred to as “a 10-bit A/D instrument”.

The chart in Figure 33 helps to demonstrate the difference between an 8-bit and a 12-bit data collector. Eight bit (8-bit) data collectors have some serious shortcomings about which an analyst needs to be aware. The dynamic range of an 8-bit instrument is about 48 dB, whereas the dynamic range for a 12-bit instrument is approximately 72 dB, as shown in the figure. The difference in dynamic range is 24 dB; that is, the 12-bit instrument is 24 dB more sensitive. What does this mean? How much more sensitive does that make the 12-bit instrument than the 8-bit instrument?

By using the formula provided in Figure 33, one could find the antilog of (dB/20), which simply involves dividing 24 dB by 20 which is equal to 1.2 ($10^{1.2}$). Then, using a calculator, hit the “INV” or “2ndF” and the “LOG” key; enter in 1.2, and hit the “ = ” key. The answer is 15.85 or 16. This means that the 12-bit A/D instruments are 16 times more sensitive than the 8-bit instruments. Being more sensitive means that it is able to much more readily respond to or record small changes or variations. Thus, the more sensitive an instrument is, the more likely it will display small peaks that are sidebands and harmonics of fundamental frequencies. Not only will they be displayed, but the amplitude and frequency values will be more accurate.

To illustrate the difference between an 8-bit and 12-bit instrument, we can examine what happened on a forming fan at a major fiberglass manufacturing client when measurements were simultaneously taken with both an 8-bit instrument (IRD 818) and a 12-bit instrument (IRD 890) on the same bearing housing, with the same IRD 970 accelerometer connected to each instrument (initially connected to the 890; then the 970 was left in place, but the connector at the data collector was removed from the 890 and attached to the 818). Note that the cascade spectrum in Figure 34 shows that measurements had been taken for several surveys with the IRD 818. Then, when Technical Associates visited the plant with an IRD 890, an abruptly different signature was noticed by our analyst as per the 12/3/90 spectrum in Figure 34. This was reported to the plant and a decision was made to conduct the simultaneous measurement test at first opportunity during the seminar.

On 12/5/90, this simultaneous measurement test was performed. First the 12-bit IRD 890 captured the data at 13:51; then the transducer cable connector was removed from the 890 and connected to the 818. The 818 captured the spectrum at 13:52 which is also shown in Figure 34. The important difference between the spectra captured by the 8-bit and 12-bit instrument is the presence of several families of 1X RPM sidebands around the inner race defect frequency (BPFI) located at 8.18X RPM in the IRD 890 spectrum, compared to their total absence in the IRD 818 spectrum. Note that when multiple sideband frequencies appear around a bearing defect frequency, they can indicate further bearing deterioration and/or amplitude modulation of BPFI as the defect passes in and out of the load zone. In the case of the 12-bit spectrum on Figure 34, note the presence not only of a single pair of sidebands, but 2 fully developed sideband families, with a third family on the way. Although this 12-bit spectrum does not indicate imminent failure, particularly due tot he absence of any other bearing frequencies or bearing frequency harmonics, the multiple families of sidebands that were totally missed by the 8-bit instrument suggests that this machine deserves close examination.

A similar test was performed on another machine at another time comparing an 8-bit and a 14-bit instrument. In this case, Figure 35 shows the spectrum from an 8-bit collector (SKF/Palomar 6100A), while Figure 36 shows the plot produced by a 14-bit collector (SKF/Palomar 6101A),

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8 BIT VERSUS 12 BIT INSTRUMENTS

DECIBEL CONVERSION TABLE

$$dB = 20 \text{ LOG} \frac{V_i}{V_o}$$

$$\frac{V_i}{V_o} = \text{ANTILOG}\left(\frac{dB}{20}\right)$$

dB	$\frac{V_i}{V_o}$ Ratio	Dynamic Range # A/D BITS used	# of bits Value	Largest Number	dB Range
100	100,000:1	1	1	2	6
90	31,623:1	2	2	4	12
80	10,000:1	3	4	8	18
70	3,162:1	4	8	16	24
60	1,000:1	5	16	32	30
50	316:1	6	32	64	36
48	251:1	7	64	128	42
46	200:1	8	128	256	48
40	100:1	9	256	512	54
34	50:1	10	512	1024	60
26	20:1	11	1024	2048	66
20	10:1	12	2048	4096	72
14	5:1	13	4096	8192	78
12	4:1	14	8192	16,384	84
10	3.16:1	15	16,384	32,768	90
8	2.51:1	16	32,768	65,536	96
6	2:1				
5	1.78:1				
4	1.58:1				
3	1.41:1				
2	1.26:1				
1	1.12:1				
0	1:1				

$\frac{4096}{256} = \frac{16}{} (24 \text{ dB})$

12 BIT A/D INSTRUMENTS ARE 16 TIMES AS SENSITIVE AS ARE 8 BIT INSTRUMENTS!

1 Part in 100 = 40dB

1 Part in 200 = 46dB **(8 BIT)**

1 Part in 1000 = 60dB

1 Part in 2000 = 66dB

1 Part in 4000 = 72dB **(12 BIT)**

FIGURE 33
CONVERSION CHART FOR NUMBER OF BITS, dB, AND LINEAR RATIO TO SHOW THE DIFFERENCE BETWEEN 8-BIT AND 12-BIT INSTRUMENTS IN DYNAMIC RANGE

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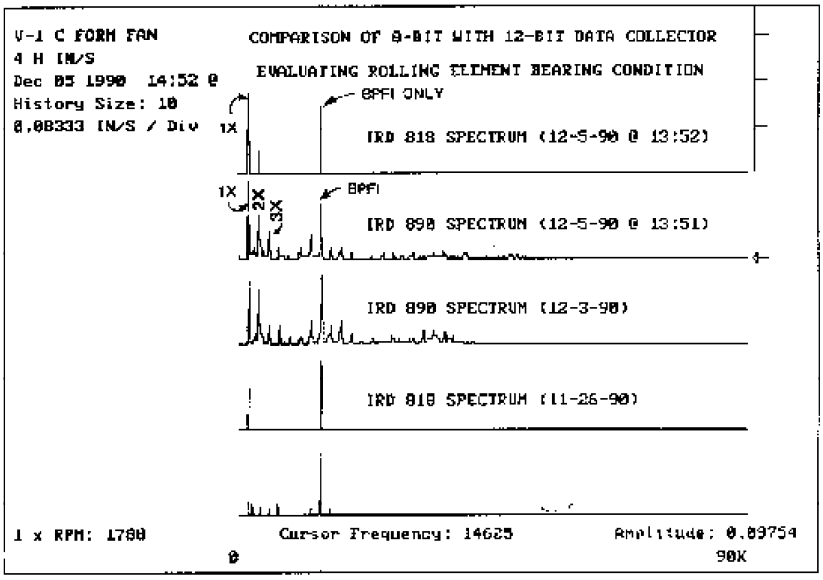

002% x 0.5000 = 0.01000 1 x RPM: 1788.00

Machine	Pos	Dir	Units	Date
V-1 C FORM FAN	4	H	IN/S	Dec 05 1990

Frequency	1 x RPM	Amplitude		Frequency	1 x RPM	Amplitude
1800	1.01	0.14249		40050	22.40	0.01001
2700	1.51	0.02002				
3600	2.01	0.07752				
5400	3.02	0.04996				
7200	4.03	0.02248				
9000	5.03	0.01001				
11025	6.17	0.02002	← BPFI - 2X			
12825	7.17	0.04750	← BPFI - 1X			
14625	8.18	0.09754	← BPFI			
16425	9.19	0.02248	← BPFI + 1X			
18225	10.19	0.02248	← BPFI + 2X			
20025	11.20	0.01247	← BPFI + 3X			
25650	14.35	0.01001				
27450	15.35	0.01001				
30825	17.24	0.01247				
34425	19.25	0.01247				
36225	20.26	0.01501				

DETECTED BY 12-BIT IRD 890

T - Threshold adjust Q - Quit

FIGURE 34
COMPARISON OF AN 8-BIT WITH A 12-BIT DATA COLLECTOR TO EVALUATE ROLLING ELEMENT BEARING CONDITION

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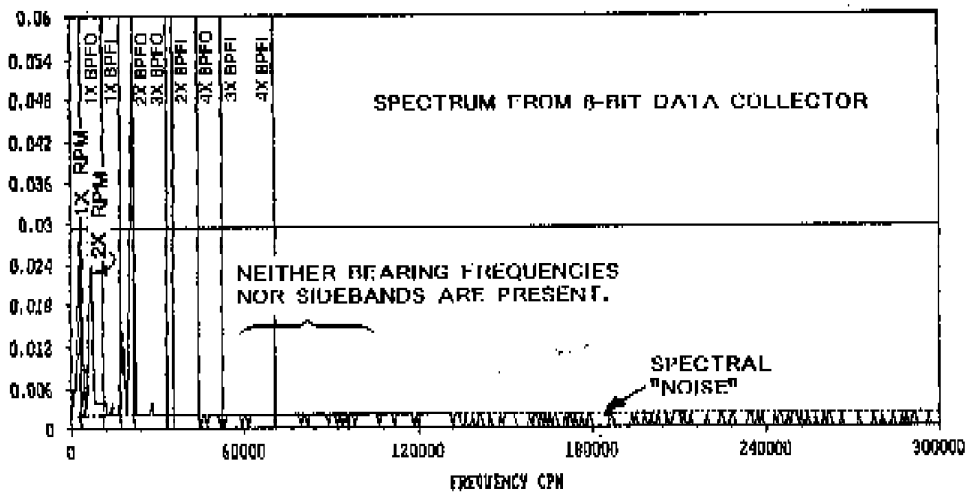
16-OCT-91 P/M FIBER SUPPLY P/M Page 1

SINGLE SPECTRUM PLOT

SET: M-806 TYPE: FFT
POINT ID: M-806 MT-0BVV DESC:
WINDOW: HANNING LINES: 400 AVER: 3 FREQ: 0 - 300000 CPM
DETECT: PEAK RPM: 3575 THRESHOLD: 0.0020 UNITS: IPS

FREQ: 3750.00 AMP: 0.0292 ORDER: 1.049 DEG: ---

SPECTRUM FROM SKF/PALOMAR 6100A (8-BIT A/D)



IDENTIFICATION OF SPECTRAL PEAKS ABOVE THRESHOLD

NO.	AMP.	FREQ.	ORDER	NO.	AMP.	FREQ.	ORDER
1.	0.0293	3750.0	1.049				
2.	0.0234	7500.0	2.098				
3.	0.0039	12000.0	3.357				
4.	0.0039	14250.0	3.986				
5.	0.0137	18000.0	5.035				
6.	0.0527	21000.0	5.874				
7.	0.0039	27750.0	7.762				

SPECTRAL ENERGY SUMMARY

OVERALL	SYNC	SUBSYNC	NONSYNC
0.0733	0.06888	0.005751	0.0244

IDENTIFICATION OF FREQUENCY DATA

ID	LABEL	AMPL	FREQ
CUSTOM NTN 6314Z BPFO	1*BPFO	0.0039	11250.00
CUSTOM NTN 6314Z BPFI	1*BPFI	0.0078	17250.00
CUSTOM NTN 6314Z BSF	1*BSF	0.0234	7500.00
CUSTOM NTN 6314Z FTF	1*FTF	0.0059	1500.00
CUSTOM NTN 6314Z BPFO	2*BPFO	0.0332	21750.00
CUSTOM NTN 6314Z BPFI	2*BPFI	0.0020	35250.00
CUSTOM NTN 6314Z BSF	2*BSF	0.0020	15000.00
CUSTOM NTN 6314Z FTF	2*FTF	0.0215	3000.00
CUSTOM NTN 6314Z BPFO	3*BPFO	0.0020	33000.00
CUSTOM NTN 6314Z BPFI	3*BPFI	0.0020	52500.00
CUSTOM NTN 6314Z BSF	3*BSF	0.0332	21750.00

16-OCT-91 P/M FIBER SUPPLY P/M Page 2

CUSTOM NTN 6314Z FTF	3*FTF	0.0117	4500.00
CUSTOM NTN 6314Z BPFO	4*BPFO	0.0020	44250.00
CUSTOM NTN 6314Z BPFI	4*BPFI	0.0020	70500.00
CUSTOM NTN 6314Z BSF	4*BSF	0.0020	29250.00
CUSTOM NTN 6314Z FTF	4*FTF	0.0020	5250.00

FIGURE 35
SPECTRUM FROM AN SKF/PALOMAR 6100A (8-BIT A/D)

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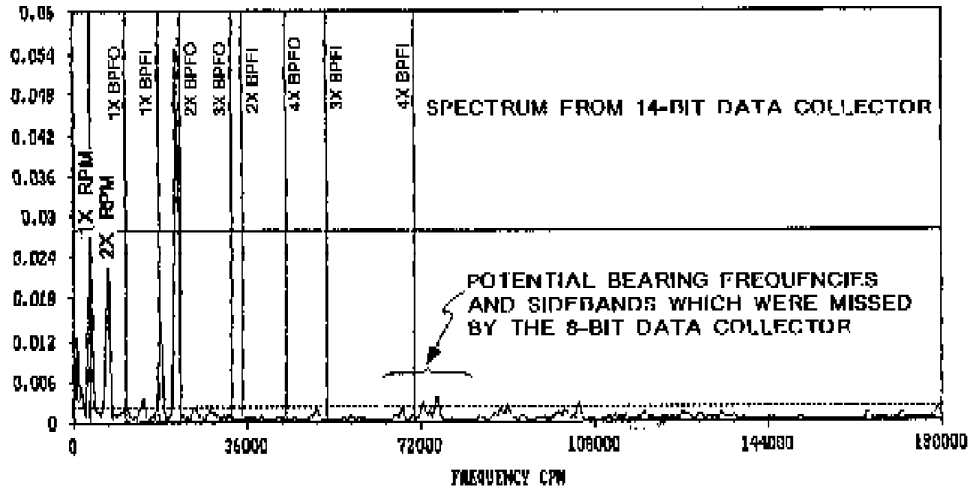
16 OCT 91 P/M FIBER SUPPLY P/M Page 1

SINGLE SPECTRUM PLOT

SET: M-806 TYPE: FFT
POINT ID: M-806 MT-QBVV DESC:
WINDOW: HANNING LINES: 400 AVER: 3 FREQ: 0 - 180000 CPM
DETECT: PEAK RPM: 3575 THRESHOLD: 0.0020 UNITS: IPS

FREQ: 3600.00 AMP: 0.0277 ORDER: 1.007 DEG: ---

SPECTRUM FROM SKF/PALOMAR 6101A (14-BIT A/D)



IDENTIFICATION OF SPECTRAL PEAKS ABOVE THRESHOLD

NO.	AMP.	FREQ.	ORDER	NO.	AMP.	FREQ.	ORDER
1.	0.0122	900.0	0.252	9.	0.0021	25200.0	7.049
2.	0.0048	1800.0	0.503	10.	0.0020	50400.0	14.098
3.	0.0278	3600.0	1.007	11.	0.0027	72450.0	20.266
4.	0.0221	7200.0	2.014	12.	0.0037	75150.0	21.021
5.	0.0021	11250.0	3.147	13.	0.0020	88650.0	24.797
6.	0.0033	14400.0	4.028	14.	0.0022	90000.0	25.175
7.	0.0139	18000.0	5.035	15.	0.0026	104850.0	29.329
8.	0.0541	21150.0	5.916				

SPECTRAL ENERGY SUMMARY

OVERALL	SYNC	SUBSYNC	NONSYNC
0.0687	0.06293	0.0111	0.02523

IDENTIFICATION OF FREQUENCY DATA

ID	LABEL	AMPL	FREQ
CUSTOM NTN 6314Z BPFO	1*BPFO	0.0019	10800.00
CUSTOM NTN 6314Z BPFI	1*BPFI	0.0105	17550.00
CUSTOM NTN 6314Z BSF	1*BSF	0.0221	7200.00
CUSTOM NTN 6314Z FTF	1*FTF	0.0029	1350.00
CUSTOM NTN 6314Z BPFO	2*BPFO	0.0014	22050.00
CUSTOM NTN 6314Z BPFI	2*BPFI	0.0002	35100.00
CUSTOM NTN 6314Z BSF	2*BSF	0.0014	14850.00
CUSTOM NTN 6314Z FTF	2*FTF	0.0009	2700.00
CUSTOM NTN 6314Z BPFO	3*BPFO	0.0004	32850.00
CUSTOM NTN 6314Z BPFI	3*BPFI	0.0002	52650.00

16-OCT-91 P/M FIBER SUPPLY P/M Page 2

CUSTOM NTN 6314Z BSF	3*BSF	0.0014	22050.00
CUSTOM NTN 6314Z FTF	3*FTF	0.0121	4050.00
CUSTOM NTN 6314Z BPFO	4*BPFO	0.0001	44100.00
CUSTOM NTN 6314Z BPFI	4*BPFI	0.0004	70650.00
CUSTOM NTN 6314Z BSF	4*BSF	0.0011	29250.00
CUSTOM NTN 6314Z FTF	4*FTF	0.0011	5400.00

FIGURE 36
SPECTRUM FROM AN SKF/PALOMAR 6101A (14-BIT A/D)

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again connecting the identical accelerometer to each analyzer. Note from Figure 36 that 14-bit A/D instruments have a theoretical dynamic range of 84 dB which would make them 64 times the amplitude sensitivity of 8-bit models (SKF/Palomar specifications show an 80 dB dynamic range which would make the 6101A approximately 40X as sensitive as its 6100A counterpart). Looking at Figure 35 note particularly the bearing frequency harmonic and accompanying sidebands at about 72,000 CPM in Figure 36 (captured by the 14-bit collector) which again are totally absent from the 8-bit spectrum of Figure 35. In addition, note the elevated "noise floor" of the 8-bit spectrum which appeared at about .0025 in/sec when this spectrum was set to the same full scale level of .06 in/sec as was the 14-bit spectrum. Importantly, it has been personally witnessed that bearings have had considerable damage even though they had defect frequency amplitudes even lower than this .0025 in/sec when they had a number of bearing frequency harmonics accompanied by multiple sideband families present in their spectra.

Therefore, it appears that 8-bit data collectors having a dynamic range on the order of 46 to 48 dB are inadequate for dependably detecting potentially serious bearing, gear, electrical and hydraulic flow problems in today's industrial machinery and machine tools due to their amplitude insensitivity/dynamic range problems, particularly where high amplitudes at 1X or 2X RPM coexist with low level (but important) frequencies and sidebands. An analyst should instead employ FFT instruments with a minimum of 12 A/D bits and a 70 to 72 dB dynamic range.

3.8 WHAT IS OVERALL VIBRATION?

Overall vibration is different from vibration at specific frequencies as discussed thus far in that it is a total summation of all the vibration, with no regard to any particular frequency. There are two types of overall vibration levels that an analyst can decide on using: DIGITAL Overall Level and ANALOG Overall Level. The Analog Overall Level is limited by the vibration instrument and transducer performance characteristics used, whereas the Digital Overall Level is limited primarily by the frequency range (0 - F_{MAX}) specified by the analyst. The advantage of specifying an Analog Overall option over a Digital Overall will be discussed in this section. To begin with, the Digital Overall Level will be considered first.

3.81 DIGITAL (OR SPECTRAL) OVERALL LEVEL

The vibration seen in a spectrum is the total sum of all the vibration measured by the transducer within the frequency span chosen. The transducer may even sense vibration from an adjacent machine and include it within the vibration spectrum. Whatever is "seen" by the transducer, regardless of its source, becomes part of the spectrum and the overall vibration.

On the latest equipment, this spectral overall is calculated automatically either within the analyzer and/or back within the host software of the computer. However, not every analyst is equipped with the latest equipment. Older instruments and software often do not have this capability. Nevertheless, there are two techniques that an analyst can use to determine the spectral or "Digital Overall". One technique employs an exact equation and the other technique uses an approximation formula with a correction factor. Figure 37 shows the exact equation that can be used for calculating the digital overall vibration level of a spectrum. This is known as the "RSS" level, meaning "Root Sum Square" level as defined in the equation on Figure 37.

To calculate the exact Digital Overall Vibration Level of a spectrum, take the amplitude of each frequency bin (A_i) and square it $(A_i)^2$, add all the squared amplitudes together; take the square root of that sum: and divide this sum by the noise factor for the FFT Window chosen (the Hanning Window is used most for PMP route measurements and has a 1.5 noise factor). Of course, this is a very lengthy process for a 400-line spectrum (and even more so for a 3200-line FFT). However, this merely illustrates what the latest equipment will do automatically; it is rarely necessary to do it manually today.

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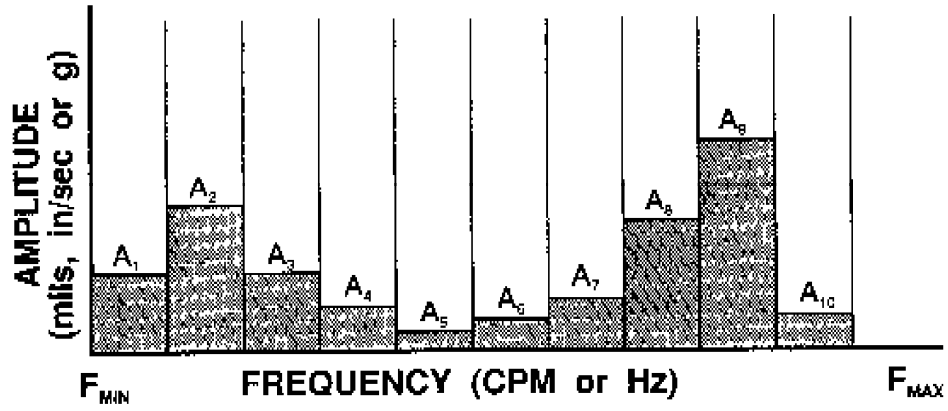

$$OA_{RSS} = \frac{\sqrt{\sum_{i=1}^{n} A_i^2}}{\sqrt{N_{BF}}} = \frac{\sqrt{A_1^2 + A_2^2 + A_3^2 + \ldots + A_n^2}}{\sqrt{1.5}}$$

OA_{RSS} = Digital Overall Level of Vibration Spectrum.
RSS = Root Sum Square Level which is the "Summation of Energy" in the spectrum from F_{MIN} to F_{MAX}.
n = Number of FFT Lines of Resolution.
A_i = Amplitude of Each of the FFT Lines.
N_{BF} = Noise Bandwidth for Window Chosen 1.5 for Hanning Window.

Which means $OA_{RSS} = .8165\sqrt{\sum_{i=1}^{n} A_i^2}$

FIGURE 37
EXACT EQUATION FOR CALCULATING DIGITAL OVERALL LEVEL OF A SPECTRUM

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Figure 38 shows an approximate formula for calculating the overall vibration spectrum and a sample spectrum showing how this approximation is applied. Here, the amplitudes from 5 separate frequency peaks were combined to compute the approximate overall level of the spectrum. Figure 38 also illustrates a problem with relying on an increase in overall vibration levels to indicate a worsening condition; in this case, a deteriorating bearing. The baseline spectrum showed that a bearing frequency had an amplitude of .030 in/sec. However, four weeks later, the spectrum taken showed that the bearing frequency peak had increased to .120 in/sec. The peak was now 4X larger. Notice the calculated overall vibration levels that were determined from the approximate equation. At baseline, the overall vibration level was .435 in/sec. Four weeks later, the overall vibration was calculated to be .450 in/sec. If one would have only relied on overall vibration levels, this slight increase in value would be no cause for alarm. However, there is cause for alarm. A spectral peak increased 4 times in value in just 4 weeks! The point is, whether the overall vibration level is calculated or measured is not significant; it will not adequately indicate a change in spectral information. A spectrum will show minute, as well as large changes in the spectral peaks and/or the amplitudes.

Figure 39 is a sample of an estimated overall vibration calculation using an actual spectrum. Note that the estimated overall value that was calculated is .161 in/sec. As a comparison, the actual overall, as measured by the data collector, was .182 in/sec. The actual overall is always higher than the estimated method since it includes all the individual amplitudes within each bin, not only the detectable peaks over the user-defined threshold. (An example of a user-defined threshold can be seen in Figure 16). By multiplying the estimated value by a 1.1 correction factor, derived experimentally, the estimated vibration will be fairly close to the actual (i.e., .161 X 1.1 = .177 in/sec; very close to the actual .182 in/sec overall vibration). Fortunately, the data collectors and computer software perform all these calculations and one only needs to do this calculation under unusual circumstances.

3.82 ANALOG OVERALL LEVEL

One of the problems of looking at only the Digital (or Spectral) Overall is that any significant vibration occurring outside the frequency range (0 - F_{MAX}) specified by the analyst will not be "seen". For example, if a high level of .60 in/sec was occurring out at a frequency of 100,000 CPM and an F_{MAX} of only 60,000 CPM was chosen, this high-amplitude 100,000 CPM frequency would not be included within the Digital Overall. In this case, the Digital Overall might only be calculated as .20 in/sec. This would be far below the Digital Overall value that could be obtained if a frequency range out to 120,000 CPM had been chosen.

To overcome this problem, some PMP systems determine the overall by looking directly at the time waveform over a very wide fixed frequency range. Then, the overall measured by the analyzer will be totally independent of any frequency-span spectrum chosen by the user. This is what is known as the Analog Overall. For example, one system uses a time waveform having a fixed frequency span from 300 to 3,900,000 CPM (5 to 65,000 Hz). In the above example (with the .60 in/sec at 100,000 CPM), the Analog Overall would be dramatically higher than the Spectrum/Digital Overall where an F_{MAX} of only 60,000 CPM was chosen. For this reason, if the PMP system allows the option of finding either the Spectrum (or Digital) Overall or Analog Overall, choose the Analog Overall option.

By comparing the differences in value between the Analog Overall and the Digital Overall, the analyst will be able to determine if there is more spectral content that is beyond the F_{MAX} than was specified. If there is, he can rectify the situation by specifying a larger F_{MAX} (perhaps to the maximum) in order to capture that data. Regardless of which Overall Level measuring system is used, a spectrum is a better indicator of the actual change in machine condition. The Analog Overall Level merely lets the analyst know if his F_{MAX} is set high enough to capture all the spectral content.

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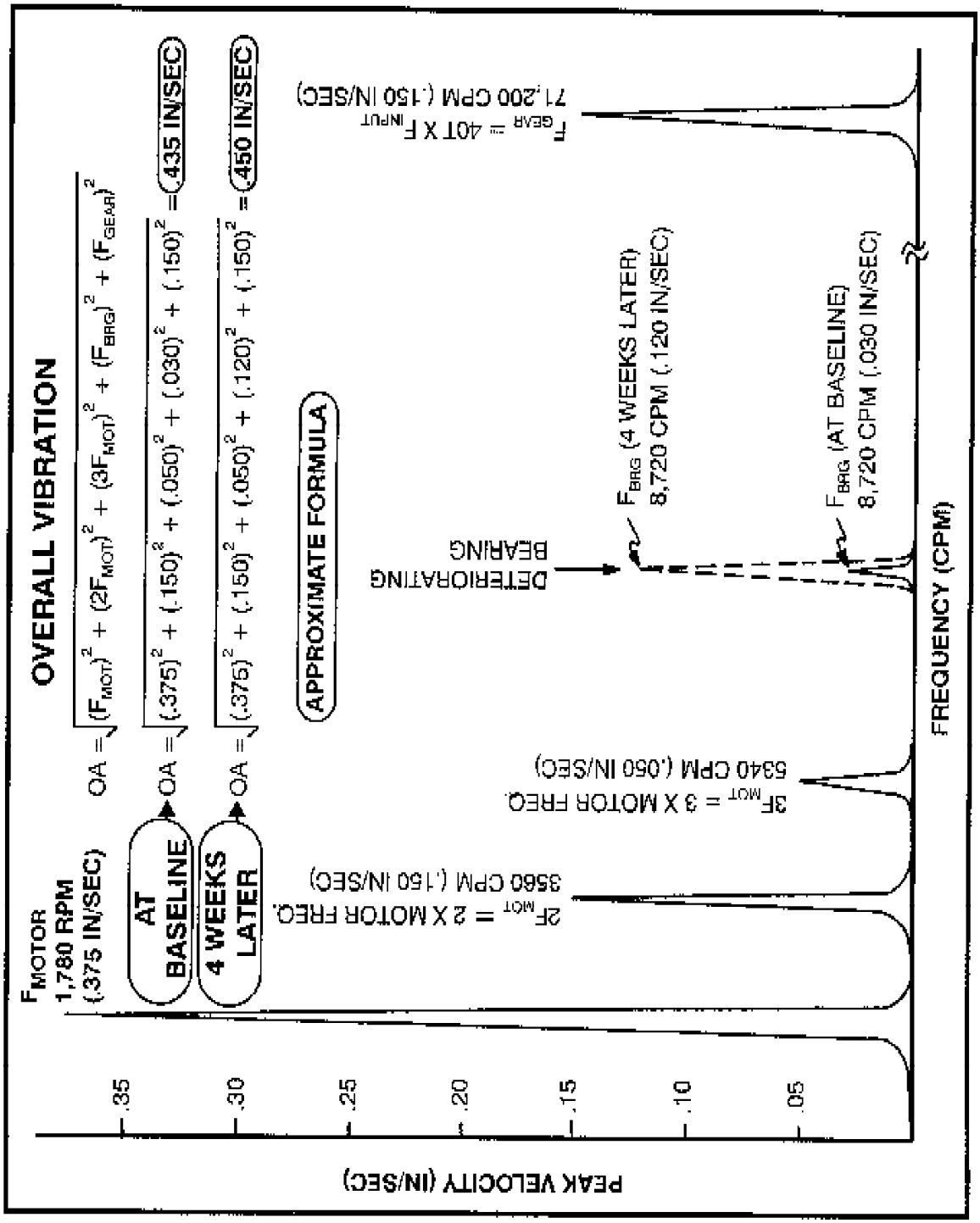
OVERALL VIBRATION
$OA = \sqrt{(F_{MOT})^2 + (2F_{MOT})^2 + (3F_{MOT})^2 + (F_{BRG})^2 + (F_{GEAR})^2}$
$OA = \sqrt{(.375)^2 + (.150)^2 + (.050)^2 + (.030)^2 + (.150)^2} = 435$ IN/SEC
$OA = \sqrt{(.375)^2 + (.150)^2 + (.050)^2 + (.120)^2 + (.150)^2} = 450$ IN/SEC
APPROXIMATE FORMULA
AT BASELINE
4 WEEKS LATER
F_{MOTOR} 1,780 RPM (.375 IN/SEC)
$2F_{MOT}$ = 2 X MOTOR FREQ. 3560 CPM (.150 IN/SEC)
$3F_{MOT}$ = 3 X MOTOR FREQ. 5340 CPM (.060 IN/SEC)
F_{GEAR} = 40T X F_{INPUT} 71,200 CPM (.150 IN/SEC)
F_{BRG} (4 WEEKS LATER) 8,720 CPM (.120 IN/SEC)
F_{BRG} (AT BASELINE) 8,720 CPM (.030 IN/SEC)
DETERIORATING BEARING
PEAK VELOCITY (IN/SEC)
.35
.30
.25
.20
.15
.10
.05
FREQUENCY (CPM)

FIGURE 38
APPROXIMATE FORMULA FOR CALCULATING THE OVERALL LEVEL OF A SPECTRUM

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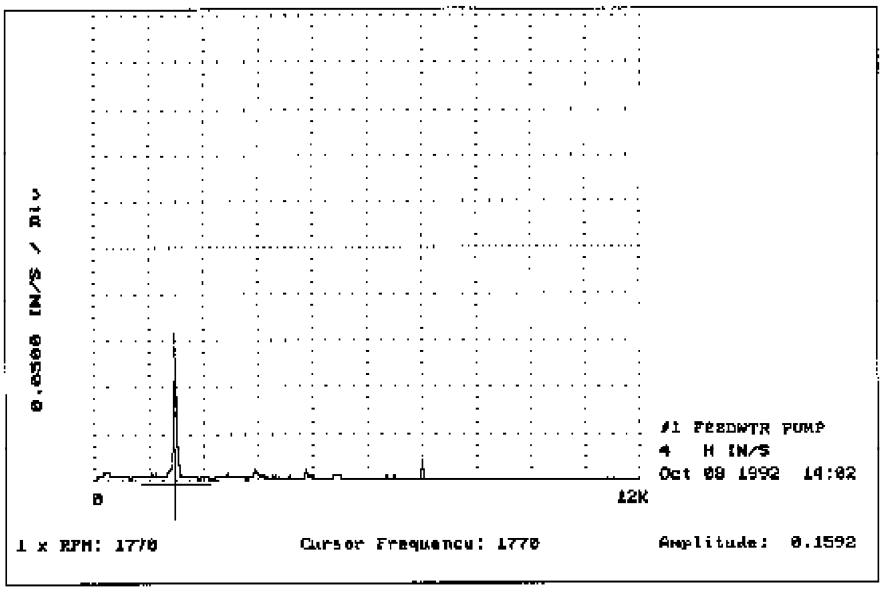

1% x 0.500 = 0.0050 1 x RPM: 1770.00

Machine	Pos	Dir	Units	Date
#1 FEEDWTR PUMP	4	H	IN/S	Oct 08 1992 14:02

Frequency	1 x RPM	Amplitude
240	0.136	0.0057
300	0.169	0.0071
1230	0.695	0.0051
1770	1.000	0.1592
2910	1.644	0.0050
3510	1.983	0.0106
4650	2.627	0.0079
7200	4.068	0.0201

where OA = Overall Level of Vibration
n = Number of FFT Lines of Resolution
Ai = Amplitude of Each of the FFT Lines

$$OA = \sqrt{\sum_{i=1}^{n} A_i^2}$$

$$OA = \sqrt{(A_1)^2 + (A_2)^2 + (A_3)^2 + (A_4)^2 + (A_5)^2 + (A_6)^2 + (A_7)^2 + (A_8)^2}$$

$$OA = \sqrt{(.0057)^2+(.0071)^2+(.0051)^2+(.1592)^2+(.0050)^2+(.0106)^2+(.0079)^2+(.0201)^2}$$

$$= \sqrt{.026}$$

OA = .161 Inch/sec
OA = .161 X 1.1 (CORRECTION FACTOR)
OA = .177 Inch/sec

T - Threshold adjust Q - Quit

FIGURE 39
SAMPLE CALCULATION OF APPROXIMATE OVERALL (OA) VIBRATION

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CHAPTER 4

INTRODUCTION TO NATURAL FREQUENCY TESTING AND INSTRUMENTATION

4.0 INTRODUCTION

Every mechanical system has a series of natural frequencies, each of which has its own damping characteristics. These natural frequencies will lie "dormant" in a system until they are excited by some external influence or forcing function. The vibration can be greatly amplified if a forcing function, such as unbalance or a blade pass frequency (# blades X RPM) happens to be within the range of a natural frequency. When such forcing frequencies coincide with (or closely approach) a natural frequency, this is the condition known as resonance. In some machines (such as reciprocating or diesel engines) which are subject to impact, several such natural frequencies can be excited for reasons which will be explained later in this paper. The effects will be particularly pronounced if the frequency is sustained over a significant time duration within a system's resonant range. Some of the effects of this amplified vibration can range from premature wear and excessive maintenance to fatigue failure or complete structural collapse (Ref. 1).

Solving vibration problems due to resonance can increase uptime and decrease maintenance, replacement, and operating costs. It is a two-part process. First, the resonant frequencies need to be identified and compared to the operating speeds of the equipment. Second, to resolve the problem, a choice has to be made between changing the operating speed or changing the resonant frequencies (or both) (Ref. 1).

This chapter will introduce the analyst to various techniques and instrumentation that can be used to test and measure natural frequencies. In particular, impact/impulse testing methods will be covered, along with coastdown and runup tests as well as Bode' plot and Polar plot acquisition and interpretation.

4.1 DIFFERENCE BETWEEN NATURAL FREQUENCY, RESONANCE AND CRITICAL SPEED

The terms "natural frequency", "resonance" and "critical speed" are often used interchangeably, even by experienced vibration analysts. However, there are distinctions between each of these three terms. For example, a machine may be operating at a resonant frequency, but this may or may not be a critical speed. It is hoped the following definitions will help clarify the differences between each of these important terms.

a. Natural Frequency:

Reference 2 defines a natural frequency as "The frequency of free vibration of a system. The frequency at which an undamped system with a single degree of freedom will oscillate upon momentary displacement from its rest position." In addition, for a multiple degree-of-freedom system, the natural frequencies are the frequencies of the normal modes of vibration. *All machines and all structures have a number of natural frequencies. If forced to vibrate at one or more of these natural frequencies, dynamic stresses of 10 to 100X higher than those are induced* compared to those which would be generated if these same forces were input at other frequencies lower or higher than these natural frequencies. However, just because a machine may

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have many natural frequencies is not necessarily a problem. It only becomes a problem if they are excited (see "resonance" below). For example, when we perform impulse tests, we are not looking for resonant frequencies, but for natural frequencies which the machine must avoid so that it may not encounter resonance.

b. Resonance:

Resonance is the condition which occurs when such forcing frequencies do in fact coincide with one or more natural frequencies. These may be a natural frequencies of the rotor, but often can be a natural frequency of the support frame, foundation or even of drive belts. The piping on compressors can be a problem if they resonate. Forcing frequencies include those from sources such as unbalance, misalignment, looseness, bearing defects, gear defects, belt wear, etc. Therefore, it is important to point out resonances can be encountered *not only at 1X RPM,* but at series of frequencies including 1X RPM, 2X RPM, 3X RPM, etc. Reference 2 defines resonance as "the condition of vibration amplitude and phase change response caused by a corresponding system sensitivity to a particular forcing frequency. A resonance is typically identified by a substantial amplitude increase, and related phase shift."

c. Critical Speed:

Reference 3 points out that "*critical speeds are a special case of resonance in which the vibrating forces are caused by the rotation of the rotor*". He also adds that "*critical speed testing is often more complicated than resonance testing because the natural frequencies encountered are functions of stiffness and mass which may be dependent on machine speed*". For example, *in the case of a machine with an extended overhung rotor or in the case of general sleeve bearing machines, critical speeds can change their frequency location due to changes in stiffness and/or gyroscopic motion (unlike resonant frequencies for frames, foundations and rolling element bearing machines which have fixed natural frequencies independent of operating speed)*. Similarly, Reference 1 defines critical speeds as "in general, any rotating speed which is associated with high vibration amplitude. Often, the rotor speeds which correspond to natural frequencies of the system." Generally, good design practices mandate that a machine should be designed not to operate within ***20%*** *of a critical speed* (*not* 20% of any natural frequency).

***The point made in the last sentence of the preceding paragraph is that it is practically impossible to operate at least 20% away from any natural frequency (of the rotor, support frame, foundation and attached structure)*. And, any structure has *six (6) different sets* of natural frequencies including those in the horizontal, vertical and axial directions, as well as "rocking modes" in each of 3 directions. Thus, the key will be to identify the locations of all resonances being excited, how severe the resonant vibration is, and what is the duration of time to which the rotating or stationary structure is subjected to resonance. Finally, if the machines and structures are fabricated from steel, cast iron, aluminum or any other lightly damped material, it may only be necessary to remain approximately 10% away from resonance in order to avoid resonant amplitude amplification. This step alone can easily reduce vibration from 10 to as much as 30 times.**

Figure 1 helps illustrate what happens when a machine goes into resonance. It shows a graph of magnification factor (Q) on the vertical axis versus frequency ratio (f/f_n) on the horizontal axis on the upper graph, while plotting phase lag (degrees) versus the same frequency ratio in the lower graph. Note that the frequency ratio is the ratio of forcing frequency (f) to undamped natural frequency (f_n). This forcing frequency can be caused by many sources and can occur at many different frequencies. For example, it can be caused by unbalance at 1X RPM; or by misalignment possibly at 2X RPM; or by a gear mesh frequency at the number of gear teeth X RPM; or by blade pass frequency (BPF) at the number of blades (or vanes) X RPM; or even excited by a rolling element bearing defect frequency. Figure 1 shows that *the problem occurs*

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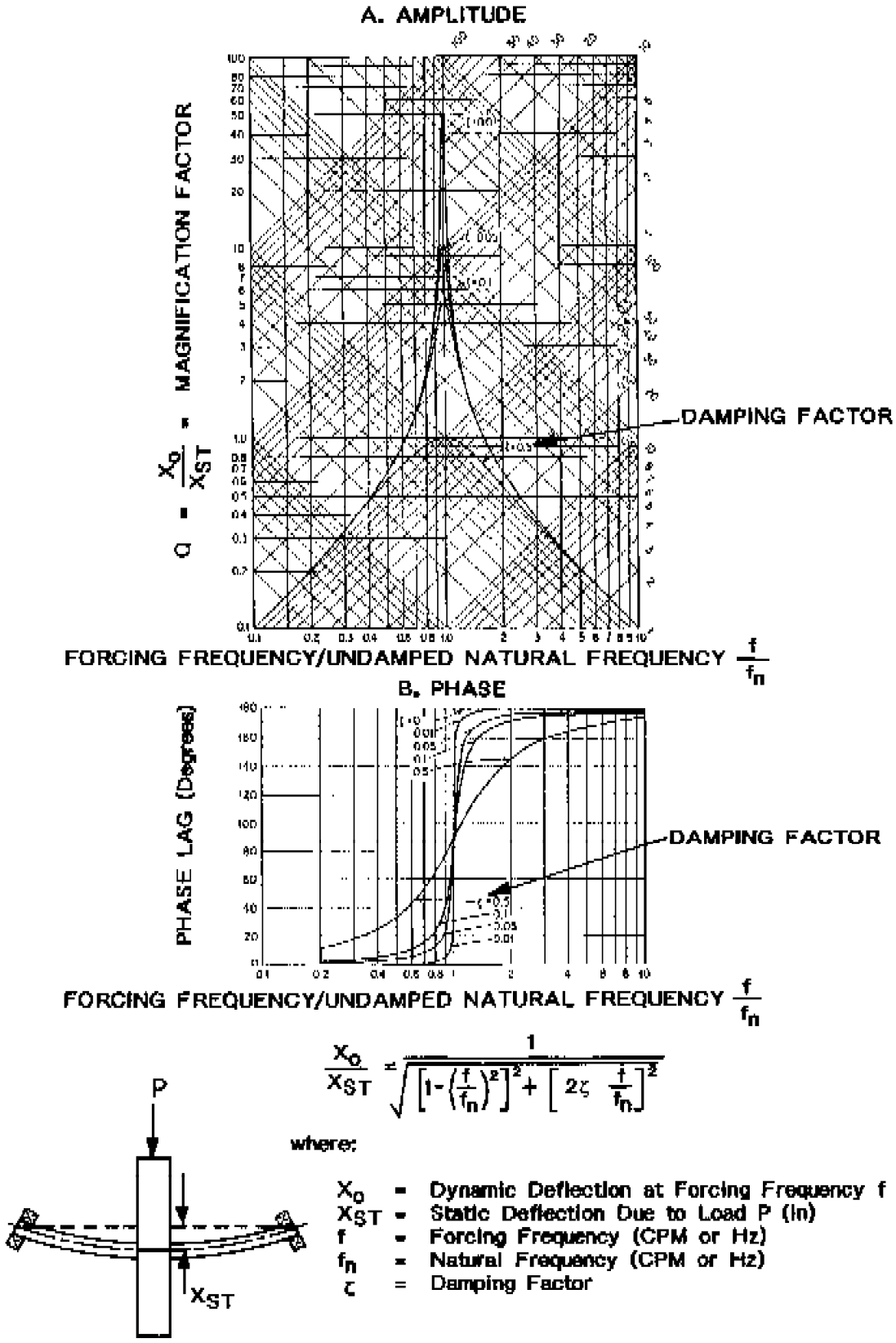
A. AMPLITUDE
Q = $\frac{X_o}{X_{ST}}$ = MAGNIFICATION FACTOR
DAMPING FACTOR
FORCING FREQUENCY/UNDAMPED NATURAL FREQUENCY $\frac{f}{f_n}$
B. PHASE
PHASE LAG (Degrees)
DAMPING FACTOR
FORCING FREQUENCY/UNDAMPED NATURAL FREQUENCY $\frac{f}{f_n}$
$\frac{X_o}{X_{ST}} = \frac{1}{\sqrt{\left[1-\left(\frac{f}{f_n}\right)^2\right]^2 + \left[2\zeta \frac{f}{f_n}\right]^2}}$
P
X_{ST}
where:
X_o = Dynamic Deflection at Forcing Frequency f (In)
X_{ST} = Static Deflection Due to Load P (In)
f = Forcing Frequency (CPM or Hz)
f_n = Natural Frequency (CPM or Hz)
ζ = Damping Factor

FIGURE 1
RESONANT AMPLIFICATION CURVE & PHASE CHANGE AT RESONANCE

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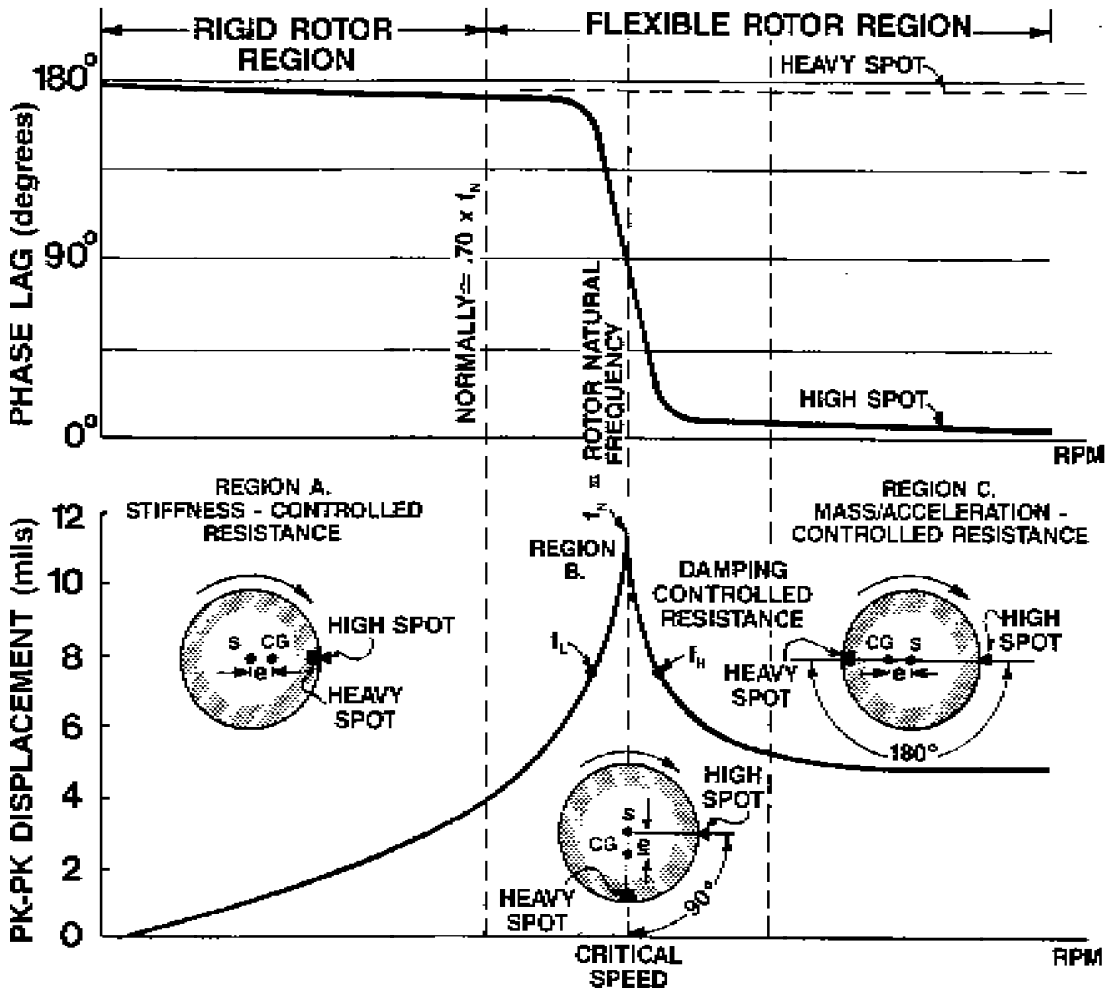

$$\frac{X_C}{X_{ST}} = \frac{1}{\sqrt{\left[1-\left(\frac{f_i}{f_N}\right)^2\right]^2 + \left[2d\frac{f_i}{f_N}\right]^2}}$$

When Forcing Freq. = Natl. Freq. ($f_i = f_N$):

$$\frac{X_Q}{X_{ST}} = \frac{1}{2d} = Q = \frac{f_N}{(f_H - f_L)}$$

WHERE:

- X_C = Dynamic Deflection at Frequency f_i (mils)
- X_{ST} = Static Deflection (mils)
- f_i = Forcing Frequency (CPM)
- f_N = Natural Frequency (CPM)
- d = Damping Factor (lb-sec/in)
- Q = Resonant Amplification Factor
- f_L = .707 A_N = Upper Half-Power Freq. (CPM)
- f_L = .707 A_N = Lower Half-Power Freq. (CPM)
- A_N = Amplitude of Natural Frequency (mils)

LEGEND

- CG = Rotor Center of Gravity
- S = Shaft Geometric Centerline
- e = Eccentricity (in.)
- Heavy Spot = Actual Unbalance Location
- High Spot = Point of Maximum Rotor Deflection

FIGURE 2
CHANGE OF VIBRATION DISPLACEMENT AND PHASE LAG WITH RPM ABOVE, BELOW, AND AT ROTOR RESONANCE

when such a forcing frequency (f) happens to equal the system natural frequency (f_n) - that is, when $f/f_n = 1.0$. When this occurs, Figure 1 shows that vibration can be amplified many times higher, often on the order of 10 or even 50 times higher vibration, depending on the amount of damping within the system *(a lightly damped system will suffer a tremendous increase when vibrating at a natural frequency).*

Therefore, *this magnification factor (Q) relates how much vibration is amplified when a machine passes through a natural frequency*. If a structure is lightly damped (damping factor = .001), Figure 1 shows that vibration can be amplified approximately *50 times higher*. In the case of most machine component materials of construction, these normally have damping factors (ζ) ranging from approximately .001 up to as high as .050. Looking at Figure 1, this means that when passing through resonance, most machines might experience an increase from 10 times up to 50 times higher vibration if the machine is allowed to dwell on the natural frequency. Therefore, such vibration can easily lead to premature, or even *catastrophic machine failure.*

Thus, resonant vibration problems can be very severe, and within the last few years, have been found to be more commonplace than ever before. Thus, the remainder of this chapter will be devoted to introducing the analyst on how to conduct some rather simple tests for natural frequencies should he suspect them to be present within his machinery, support frame, foundation, or even attached piping and building structure. Later, the "ANALYSIS III" seminar will go into much greater depth on how to not only detect natural frequency problems (using both single-channel and multi-channel analyzers), but also on how to correct them using proven, cost effective solutions.

4.2 CHANGE IN MODE SHAPE WITH HIGHER NATURAL FREQUENCIES

*A **"Mode Shape"** is the resultant deflected shape of a structure when forcing frequencies coincide with natural frequencies of the structure (i.e., condition of "resonance"). Figure 3 shows how a rotor simply supported between bearings deflects when it passes through its first, second and third critical speeds.* Note the change in mode shape as the rotor passes through its first, second and third critical speeds. Drawing A shows that the maximum deflection will be at the center of the roll when it passes through its first critical. At this point, it will have minimum displacement at each end where the bearings are located.

Drawing B of Figure 3 shows the *distinctly different mode shape as the roll enters its second critical*. Note here that the center point (mid-span) now is a point of minimum deflection (displacement). These points of minimum deflection are known as **"Nodal Points"**. A nodal point (also called a "node") are often at a point of minimum absolute shaft displacement. *Motion immediately on either side of a node is approximately 180° out-of-phase.* It is important to point out that nodal points may readily change location along the shaft axis due to changes in residual unbalance or other forcing function, or due to a change in restraint such as an increased bearing clearance.

Drawing C of Figure 3 shows the mode shape when the roll enters its third critical speed. Here, node points will again be on either end, but *two other node points will now occur at approximately one-third and two-thirds of the bearing center distance*. There will now be three *antinodes*, or points of maximum deflection each of which will occur between the bearings.

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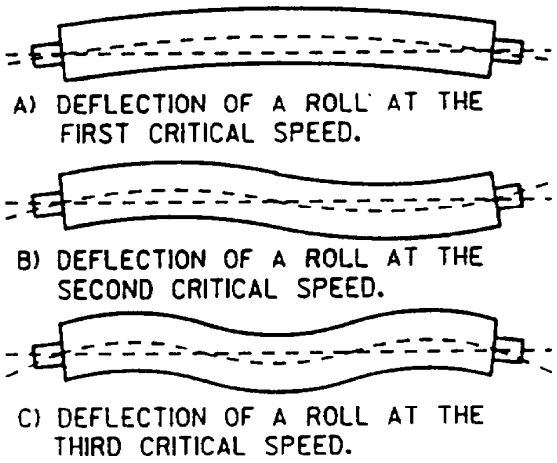
A) DEFLECTION OF A ROLL AT THE FIRST CRITICAL SPEED.
B) DEFLECTION OF A ROLL AT THE SECOND CRITICAL SPEED.
C) DEFLECTION OF A ROLL AT THE THIRD CRITICAL SPEED.

FIGURE 3
MODE SHAPES OF FIRST, SECOND AND THIRD CRITICAL SPEEDS FOR A SIMPLY SUPPORTED ROTOR

4.3 IMPACT/IMPULSE NATURAL FREQUENCY TESTING

Of all the testing techniques to measure natural frequency, impact testing is the most popular even though it might not provide the ultimate precision of the other methods discussed. For one thing, it can even be performed with fair accuracy using a single channel spectrum analyzer or data collector and a rubber mallet (or even a 4" X 4" piece of lumber). A "dead blow" or rubber mallet is generally used to repetitively tap the structure or machine. The mallet or hammer has to impact with enough force to at least produce some amplitude response so the effect can be seen. The amount of amplitude is not really significant in this test (unless one "overdrives" the input of the analyzer causing it to truncate the time waveform, thereby introducing a series of "false frequencies" and additional noise). The harder one strikes, the higher the amplitude. What really is significant is measuring the frequency or frequencies which will appear.

Use of a rubber mallet will excite all natural frequencies in the range from about 0 - 30,000 CPM (500 Hz) or higher, depending on the hardness of the rubber. Commercially available units called "modally-tuned hammers" can equally excite frequencies out to about 120,000 CPM (2000 Hz). That is, *when one of these elastomer-tipped hammers impacts the structure, it will impact roughly equal energy across a broad range of frequencies up to a certain "cut-off frequency"* where the response will drop off as shown in Figure 4. If any natural frequencies are present within this "flat response region" (Figure 4), they will be excited by the impact and will respond. Importantly, the tip material and hardness will limit the highest natural frequency which can be excited as suggested by Figure 4 (remember, do not use aluminum or steel-tipped hammers for they will corrupt the frequency response due to the "double hits" they almost surely will impart) (Ref. 4).

If a dead blow hammer or rubber mallet is not available, a four-by-four timber about five feet long is adequate for simple impact testing with a single channel analyzer. Sledge hammers with steel heads, unless covered with an elastomer, are not adequate since "double hits" or even "triple hits" again introduce noise into the frequency response. These hammers bounce off the structure and do not usually impart enough energy into the system (Ref. 5).

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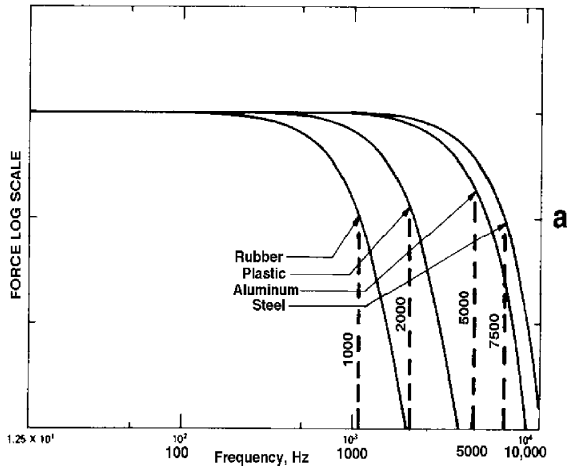

FIGURE 4
FORCE SPECTRA PRODUCED WITH VARIOUS HAMMER TIP MATERIALS

For this type of testing, older technology instruments can be used with some success. Tunable-filter analyzers (i.e., IRD 350) with a frequency meter could be used. The analyzer is placed in the "filter-out" mode during the impact. If the natural frequency is distinct, it will appear on the meter[2]. Then, as the analyzer is tuned through various frequency ranges, the natural or resonant frequencies can be located as the machine or structure is repeatedly tapped. Similarly, a swept-filter analyzer (i.e., IRD 880) could also be used for structural impact testing. The structure is *struck continuously* as the analyzer sweeps automatically through a selected frequency range[2]. Here, it will be important to attempt to impact the structure with about the same force, to allow it

FIGURE 5
IMPACT TESTS ON TWO MACHINE TOOL SPINDLES USING A SWEPT-FILTER ANALYZER

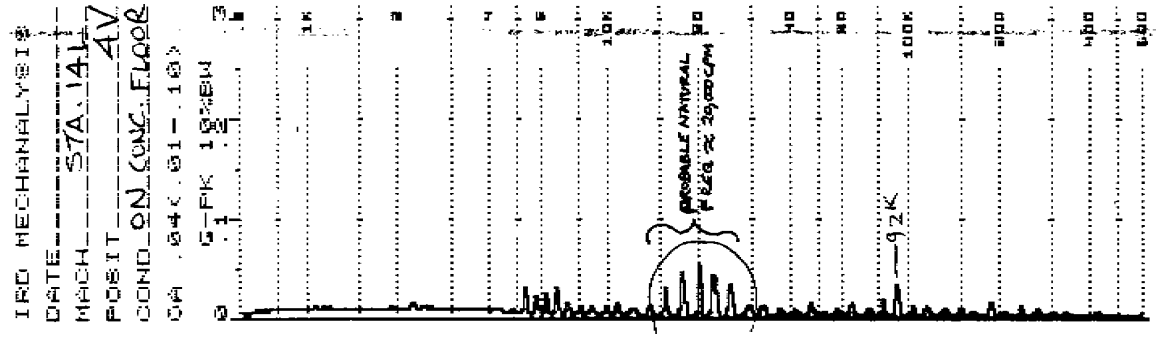

FIGURE 5A. THE NATURAL FREQUENCY IS ABOUT 20,000 CPM.

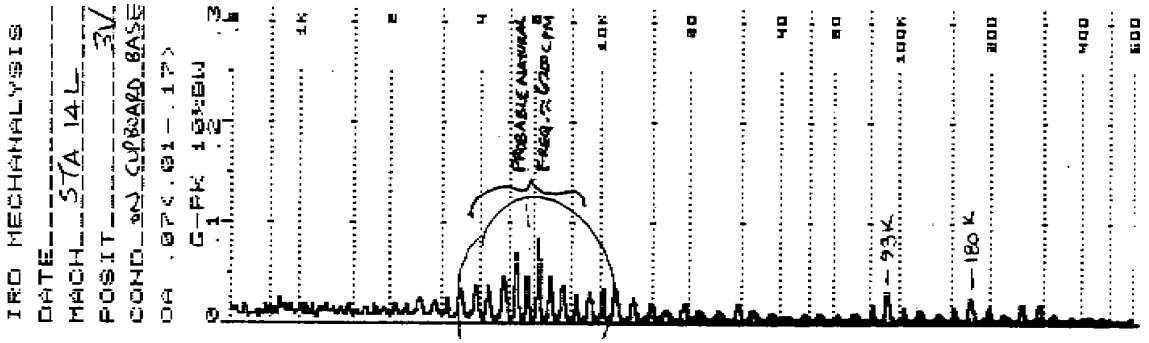

FIGURE 5B. THE NATURAL FREQUENCY IS ABOUT 6200 CPM.

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to “ring down”, time for the analyzer to slowly sweep through the frequency range, and to ensure once again one does not overdrive the analyzer electronics for valid results. In Figure 5 are amplitude versus frequency strip-charts from an IRD 880 swept-filter analyzer. A “bump” (impact) test was done on two precision machine-tool box spindles to determine their natural frequencies. A dead blow hammer was used to strike the housing in the vertical direction near the bearings. Note that in Figure 5A the natural frequency was around 20,000 CPM; and in Figure 5B the natural frequency was around 6200 CPM for that particular box spindle assembly.

However, note that the measurement conditions were different for each box spindle assembly. The one in Figure 5A was struck while it was resting on a concrete floor whereas the spindle in Figure 5B was struck while it was resting on a steel cabinet base. Although both box spindles were struck in the vertical direction, the first one was struck at the bearing nearest the tooling and the second one was struck at the bearing nearest the drive end. The point is, in order to have comparative natural frequency tests, the conditions have to be identical. The box spindles have to be identical and they have to have the same base mounting method. They also need to be impacted in the same direction with the same mallet and on the same location. If not, all the natural frequencies will be different in an impact test. That is, *the natural frequencies in the horizontal and axial directions will both be different than the natural frequencies in the vertical direction*. In addition, *the natural frequencies which will be measured will be the “system” natural frequencies,* not of any particular machine element alone. The natural frequencies measured at the drive end can be different than the natural frequencies measured at the tool end, especially if one box spindle is longer than the other.

Instead of using dead blow hammers or rubber mallets, a more sophisticated approach would be to use a “modally-tuned” hammer. Various models of modal hammers are available like the one shown in Figure 6. These can range in size from one with a .005 lb steel head for testing very light structures such as printed circuit boards; to those with 1.0 lb steel heads for general rotating machinery and medium to heavy structures; and on up to “sledge hammers” with 12 lb steel heads for exciting very heavy structures like buildings, ships and foundations. Note that a force transducer is placed just behind the hammer tip with which the impact force can be measured and the signal transmitted over to an analyzer through an electrical connector normally located at the end of the hammer’s handle. Also note that a number of tips are supplied with each hammer. These tips vary in hardness from “soft” to “hard” to excite the frequency range of interest (Ref. 6).

The modal hammer can be used in conjunction with either a single-channel or multi-channel data collector or spectrum analyzer. *In this case, it is very important to allow the structure to "Ring Down" before hitting it again.* With this analyzer, the impact tests can be performed in either the time or frequency domains. During the impact tests a uniform (no window) should be used on the analyzer. Some analyzers have special windows for impact tests (such as a force/exponential window) (Ref. 5).

Triggering can be "free run" or from a hammer source. T*riggered data capture simply means that the analyzer only collects data when the amplitude level exceeds a specified trigger level. In addition, many analyzers have the capability to automatically reject data if the electronics are over-driven by too great an impact (this prevents great corruption of the data).*

Figure 7 shows the results of an averaged impact test. In this case the analyst was measuring the principal horizontal natural frequency of a 100 HP DC motor driving a friction roll assembly through a gearbox. Four impacts with a 4" X 4" timber were averaged to reduce random signals (Ref. 3). *Note there were three responses at 38.9 Hz, 42.0 Hz and 90.4 Hz (corresponding to 2334, 2520 and 5424 CPM, respectively) which likely corresponded to the first, second and third horizontal natural frequencies.* Note that the raised “peak” on the lower, left-hand side at about 8 Hz (480 CPM) was likely only integration noise just above the lower cut-off frequency of the instrument, although it is possible this might have been close to what is known as a “rigid body mode” of the structure (which is discussed in Section 4.424).

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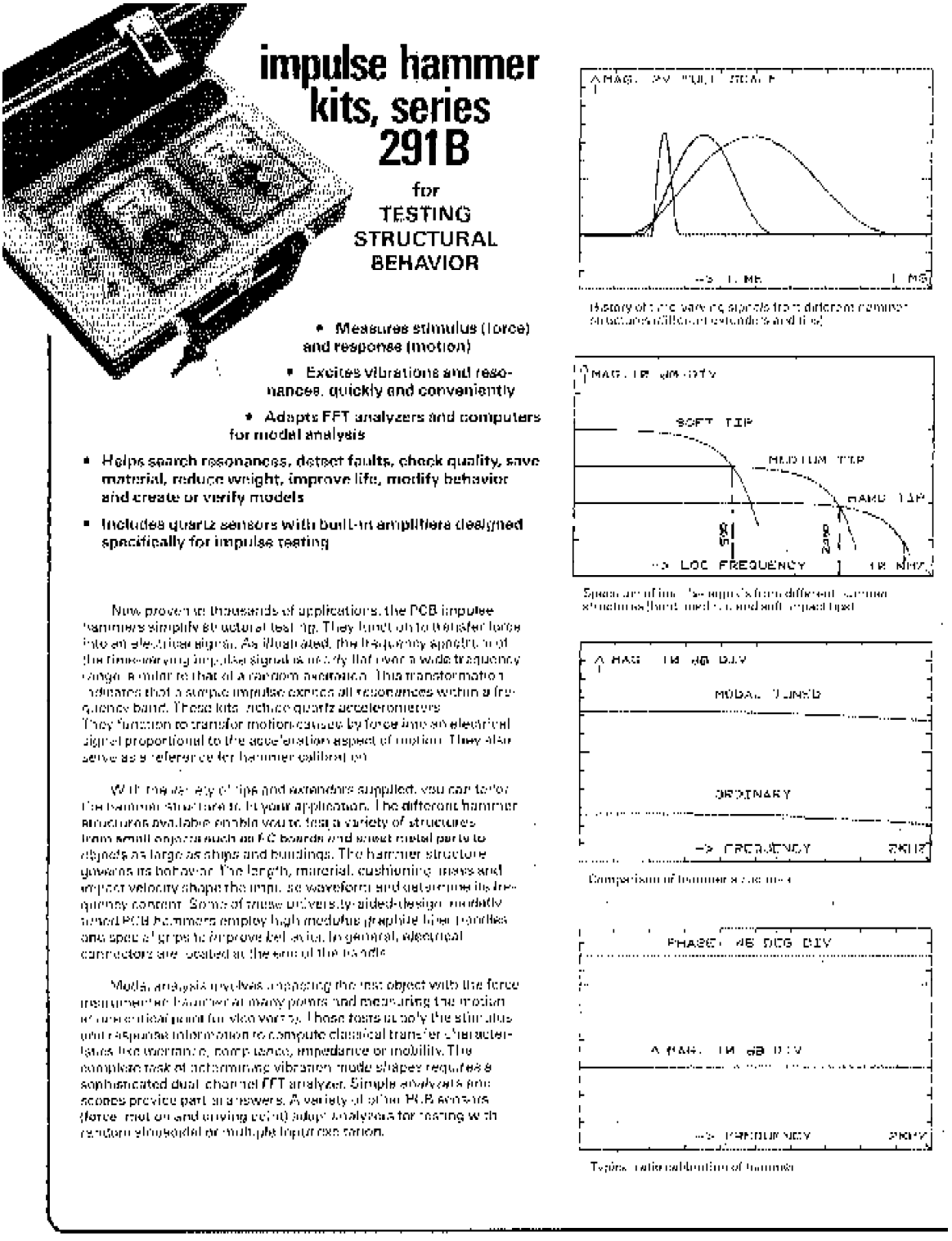
FIGURE 6
AN EXAMPLE OF A MODALLY-TUNED HAMMER (Ref. 6)


impulse hammer kits, series 291B
for TESTING STRUCTURAL BEHAVIOR
• Measures stimulus (force) and response (motion)
• Excites vibrations and resonances, quickly and conveniently
• Adapts FFT analyzers and computers for modal analysis
• Helps search resonances, detect faults, check quality, save material, reduce weight, improve life, modify behavior and create or verify models
• Includes quartz sensors with built-in amplifiers designed specifically for impulse testing
Now proven in thousands of applications, the PCB impulse hammers simplify structural testing. They function to transfer force into an electrical signal. As illustrated, the frequency spectrum of the time-varying impulse signal is nearly flat over a wide frequency range, similar to that of a random waveform. This transformation indicates that a simple impulse excites all resonances within a frequency band. These kits include quartz accelerometers. They function to transfer motion-caused by force into an electrical signal proportional to the acceleration aspect of motion. They also serve as a reference for hammer calibration.
With the variety of tips and extenders supplied, you can tailor the hammer structure to fit your application. The different hammer structures available enable you to test a variety of structures from small objects such as PC boards and sheet metal parts to objects as large as ships and buildings. The hammer structure governs its behavior. The length, material, cushioning, mass and impact velocity shape the impulse waveform and determine its frequency content. Some of these universally-aided-design modally tuned PCB hammers employ high modulus graphite fiber handles and special grips to improve feel and control. In general, electrical connectors are located at the end of the handle.
Modal analysis involves impacting the test object with the force instrumented hammer at many points and measuring the motion at one critical point for each impact. These tests supply the stimulus and response information to compute classical transfer characteristics like coherence, compliance, impedance or mobility. The complex task of determining vibration mode shapes requires a sophisticated dual channel FFT analyzer. Simple analysis and scopes provide part of the answers. A variety of other PCB sensors (force, motion and driving point) adapt analyses for testing with random, sinusoidal or multiple input excitation.
MAG. (V PEAK)
--> 1 MS
History of time varying signals from different hammer structures (different materials and soft or hard tips)
MAG. (IN UNITS)
SOFT TIP
MEDIUM TIP
HARD TIP
500
2500
--> LOG FREQUENCY
10 KHz
Spectra of impact signals from different hammer structures (hard, medium and soft impact tips)
A MAG. (IN dB D/V)
MODAL TUNED
ORDINARY
--> FREQUENCY
2KHz
Comparison of hammer structures
PHASE (45 DEG D/V)
A MAG. (IN dB D/V)
--> FREQUENCY
2KHz
Typical calibration of hammer

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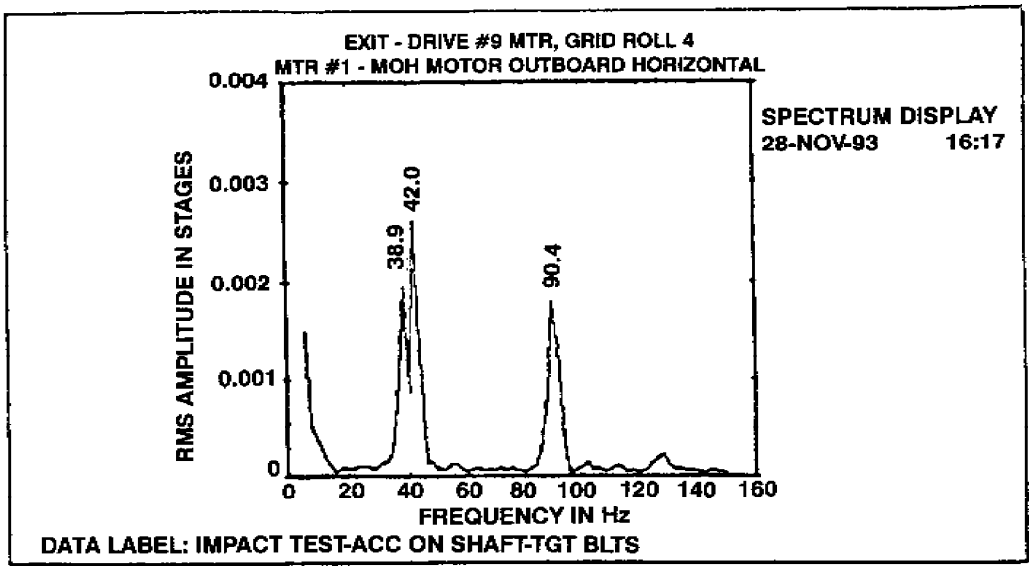

FIGURE 7
RESULTS OF AN IMPACT TEST ON A 100 HP DC MOTOR EMPLOYING AVERAGING (Ref. 7)

Averaging several impacts improves data quality, provided that a consistent impact force over the same time duration is maintained. *The single-channel analyzer cannot provide a precise estimate of the vibration response per unit load or the relative phase between response and load.* By directly measuring the force spectrum, an *estimate of average* impact force level is possible. However, without the phase between the force and response, *only crude estimates of mode shape can be made.*

A short impulse excites modes over a broad frequency range. The force pulse produced from the impact is of short duration. If it is shorter than the length of the time record, the force spectrum is quite flat at lower frequencies, as shown in Figure 8.

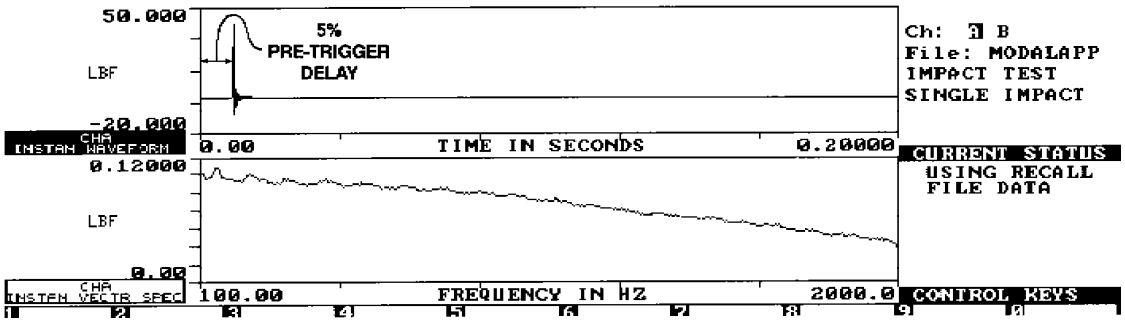

FIGURE 8
A FORCE PULSE CAUSED BY SHARP IMPACT SEEN IN TIME WAVEFORM WILL PRODUCE A FLAT FORCE SPECTRUM AT LOW FREQUENCIES (Ref. 8)

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On the other hand, Figure 9 shows the vibration response spectrum from the impact. Natural frequencies and damping can be *estimated* from the position and width of the response peaks in the spectral display. *The important thing to observe here is one needs to look at the time domain to ensure the response has returned to (or near) zero in order to ensure he has good spectral response data by minimizing leakage effects*. If it has not he may have to allow more time (t_{MAX}) for the structure to do so, or may have to employ one of the special exponential windows now available on some analyzers on the force response to force it to zero.

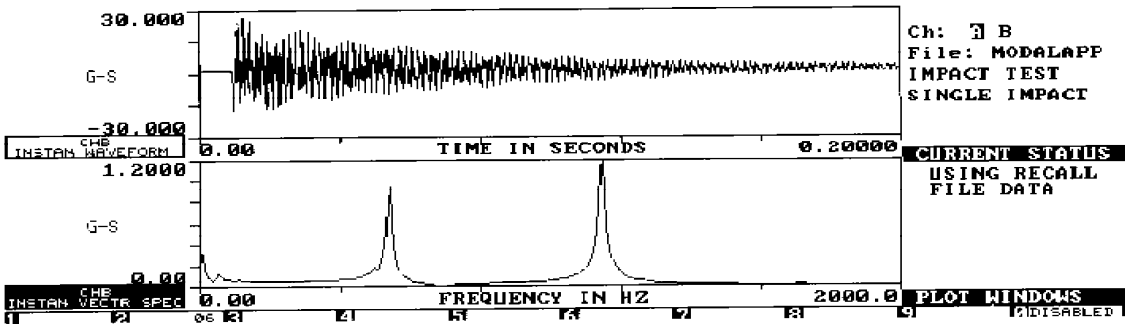

FIGURE 9
VIBRATION RESPONSE TO IMPACT SHOWING NATURAL FREQUENCIES IN THE SPECTRUM AND THE "RING-DOWN" IN THE TIME WAVEFORM (Ref. 8)

While the force signal from an impact is almost always of short duration, there may be a problem if there is background noise from surrounding machines or if there is electronic noise in the analyzer. Each will contribute to the total signal, apart from the impact, over the remainder of the time record. The signal-to-noise ratio will therefore be reduced, reducing the clarity of the test.

Impact testing produces transient signals. If a Hanning window is used, it will distort the signal since it automatically "forces" both the beginning and ending of a time block to near zero. *For correct results, a transient signal must start and finish within the time block of an FFT analysis*. As was mentioned previously, a Uniform window (no window) is the correct way to treat the signal before computing the spectrum. Smoothing windows like the Hanning or Flat Top windows are intended for continuous signals.

To ensure the impact test excites as many natural frequencies as possible, acceleration should be used rather than velocity or displacement. Higher and higher natural frequencies normally will have lower response. Acceleration data will help "elevate" them in the instrument display.

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To minimize the effect of instrument or background noise, *a force window* can be applied. It zeroes data values before and after the impact and applies a unity (1.0) weighting factor during the impact. An example force window is shown in Figure 10 which sharply filters the response just after impact. Sometimes, similar noise effects can be reduced in response analysis *by zeroing the data before the trigger* (Ref. 8).

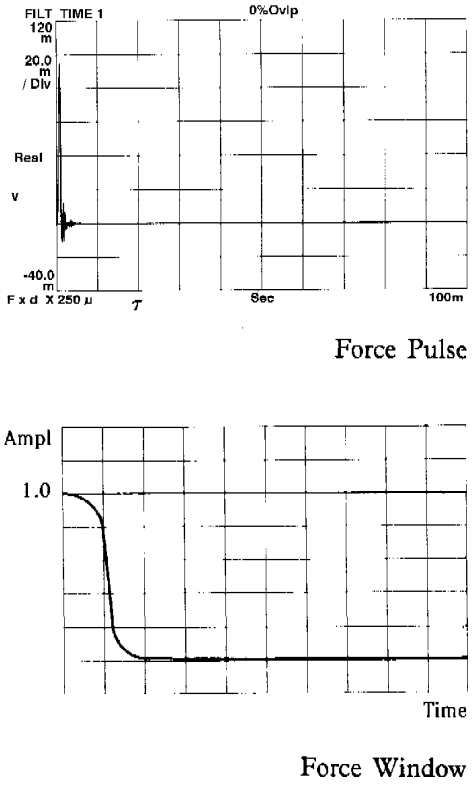

FIGURE 10
FORCE PULSE WITH WINDOW APPLIED (Ref. 9)

Apart from these effects, the vibration response signal may not always die away before the end of the time record, especially for lightly damped structures. The resulting truncation of the signal can cause significant leakage in the spectrum. This can lead to poor estimates of natural frequencies, damping, and response. To overcome this problem, *an exponential window may be used*, which applies an *exponentially decreasing* weighting to the response signal. An example exponential response window is shown in Figure 11. With sufficient weighting, the response signal *can be reduced to an insignificant level by the end of the time record*. In lightly damped systems, where the natural frequencies not affected very much by damping, exponential weighting can still be achieved effectively without affecting the natural frequencies significantly. Even for strongly damped systems, exponential weighting is sometimes useful to minimize the effects of background noise in long time records.

The setup of a Force/Exponential window combination for impact testing must be linked to the trigger delay setting, or the signals will be distorted by the windows. Pretriggering must be used, with a delay that puts the whole of the impulse waveform within the time record, equal to a small fraction of the length of the time record. A delay of 5 to 10% of the record length is sufficient. The

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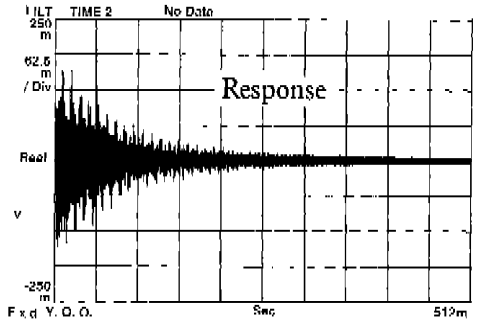
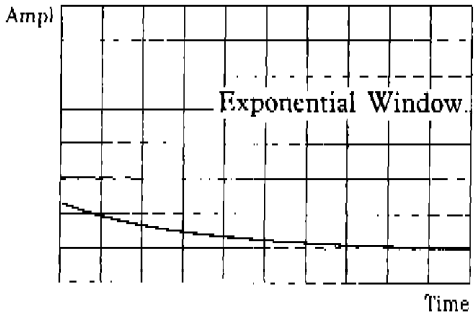
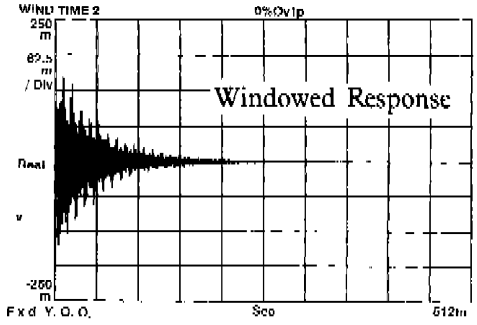


Response Continues To "Ring Down" Beyond Time Block (Not Desirable)

Exponential Decaying Response





Response Is Artificially Forced "Ring Down" By The Exponential Window

FIGURE 11
APPLICATION OF AN EXPONENTIAL WINDOW TO THE RESPONSE WAVEFORM TO FORCE IT TO GO TO ZERO BEFORE THE END OF THE TIME RECORD (Ref. 9)

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analyzer is started, causing data to be acquired continuously, until the impact force signal exceeds the trigger level. The analyzer does not stop at this point, but continues to acquire data until a complete time record block is filled. The trigger point is positioned inside the time block at the specified delay time from the beginning. Typically, the FFT is then computed, the spectrum averaged and the cycle is repeated, waiting for a new impulse to exceed the trigger level, before another spectrum is computed. The spectrum shown in Figure 9 was set up to capture the force pulse with a pre-trigger delay of 5% of the record length. The force window must begin shortly before the trigger and end shortly after the impulse. A quick test will confirm.

The exponential window can start at the same point as the force window. The exponential decay rate must be sufficient to suppress the signal by the end of the record. As a rule of thumb, the decay constant should be at least 1/4 of the length of the time record, with a short trigger delay. This reduces the signal to less than 2% of its unweighted value. Provided that there are no resonances that lie within the first 10 lines of the spectrum, there are no limitations on the frequency range or the resolution if exponential weighting is used (Ref. 4).

It was mentioned previously that averaging several impacts improves data quality *provided that* a consistent impact force over the same time duration is maintained. If one “hit” with a hammer is much “harder” than the rest, it is called “over-hitting” and will cause an overloaded force or response signal. If one “hit” is much “softer” than the rest, it is called “under-hitting” and will cause an under-ranged force or response signal. Another common problem is “double-hitting”. Typically with lightly damped structures, the structure actually rebounds before the hammer is withdrawn. This produces an uneven force level with frequency. Don't forget to allow structure to "Ring Down" before hitting it again.

All of these effects tend to corrupt the results and should be eliminated from the average. Some analyzers allow each block of data to be reviewed before being accepted, while others automatically suppress over-ranged or under-ranged data. Usually, double-hitting can be detected in the force input signal and bad data is rejected, but this is inconvenient.

In conclusion, using a swept-filter analyzer with a dead blow hammer or a single-channel analyzer with a modally-tuned hammer *does not* give precise natural frequencies. They provide reasonable, useful *estimates* of the natural frequencies. If more efficient, more precise, and more informative data is required, then a 2-channel analyzer should be used to simultaneously provide both phase and frequency information using cross-channel measurements and a coherence function.

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4.4 RUNUP AND COASTDOWN NATURAL FREQUENCY TESTS

One of the strong points about runup and coastdown tests is that they actually evaluate the machine while it is under actual running dynamic conditions. Then, *if there are any dynamic changes to natural frequencies due to gyroscopic or fluid-film pressure/stiffness changes, these will be taken into account*, particularly if the machine is allowed to come to full operating temperature. *Gyroscopic effects can sometimes be considerable on overhung rotors, particularly if the overhang is significant.* Such gyroscopic effects can cause natural frequencies to increase due to dynamic "stiffening" of the rotor.

Runup and coastdown tests can be performed with a wide array of instruments and employ a range of techniques. In essence, they can consist of nothing other than watching the overall vibration with a hand-held meter, observing how it changes as the machine coasts down from full speed. This is somewhat meaningful, but has limited useful applications. On the other hand, techniques such as Bode' plots and Polar diagrams can be applied to natural frequency testing to track and display any changes as the machine either accelerates from zero to full speed, or coasts down to a stop. Following below will be a description of how to acquire and interpret Bode' plots and Polar plots, both with and without using a tracking filter:

4.41 BODE' PLOTS

The Bode' Plot actually consists of two separate Cartesian plots - (1) Amplitude versus RPM; and (2) Phase Lag versus RPM. When a machine passes through a natural frequency during a coastdown, the Bode' Plot will indicate two changes: (1) the amplitude plot will show a peak, and (2) the phase plot will show a 180° phase shift (with a phase shift of 90° right at the natural frequency).

4.411 A Tracking Filter is Needed for Bode' Plots

A tracking filter utilizes "phase-locking" techniques to be able to record amplitude versus RPM and phase versus RPM information. If a 2-channel analyzer is used, the amplitude versus RPM plot is done on one channel while the phase versus RPM plot is done on the other channel so that each peak is displayed at an RPM which corresponds to the machine RPM at that exact moment, a reference pickup is used. The reference pickup is placed at the rotating shaft to provide a voltage pulse for each revolution of the shaft. Photocells and electromagnetic pickups are normally used to provide the reference pulse. An existing key or keyway on the shaft is an excellent target for them. If none are available, a piece of reflective tape can be used with a photocell.

The reference signal automatically tunes the tracking filter to the RPM of the shaft. If the shaft RPM changes, the frequency of the reference signal is also changed accordingly. In this way, the analyzer's filter is locked or synchronized to the shaft RPM. Also, the reference signal provides a fixed reference for comparison with the signal from the accelerometer. The result is a DC voltage proportional to the relative phase between the two signals. This DC voltage indicates a relative phase measurement (0° to 360°) on the phase versus RPM plot (Ref. 10).

4.412 Explanation of a Bode' Plot

A sample Bode' Plot is shown in Figure 12. It is apparent that the machine in question has two significant natural frequencies, one at approximately 2450 RPM and another at 4850 RPM. Of course, if there is an exciting force which corresponds to either of these frequencies, this will cause the machine to operate in resonance. These forces might be due to normal unbalance, misalignment, aerodynamic or hydraulic forces, torque pulses, reciprocating

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forces, looseness, oil whirl, etc. Then, an undesirable and probably destructive vibration can likely result. For this reason, it is helpful to know:

a) What exciting frequencies of vibration are inherent to the machine at its normal operating speed(s). These can be identified through normal amplitude-versus-frequency analysis techniques already discussed.

b) What are the natural or resonant frequencies of the machine or structure. As illustrated in Figure 12, a plot of amplitude and phase versus machine RPM will reveal these (Bode' plot).

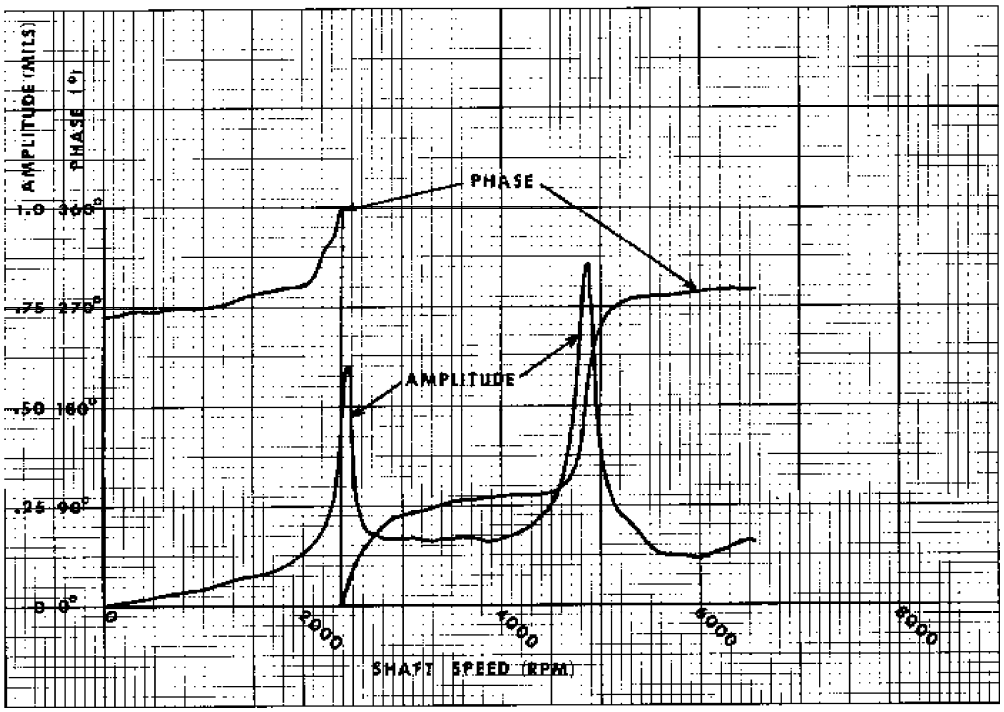

FIGURE 12
EXAMPLE BODE' PLOT USED TO CONFIRM THE LOCATION OF NATURAL FREQUENCIES (Ref. 10)

4.413 Interpreting Unusual Bode' Plots

As mentioned previously, when a machine passes through a natural frequency during coastdown or start-up, the usual indications will be a peak in amplitude at the resonance speed accompanied by the characteristic phase shift of approximately 180° (see Figure 12). Occasionally, however, the recorded data may reveal some unusual conditions of system response as illustrated by the following examples:

EXAMPLE #1:

Referring to the data in Figure 13, two distinct amplitude peaks are noted at about 500 RPM and 1200 RPM suggesting two resonant speeds. However, the recorded phase data shows the characteristic 180° phase shift for only the lower speed (500 RPM) amplitude peak. There is virtually no phase shift associated with the higher 1200 RPM peak. Based on the recorded

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data, it can be concluded that the lower 500 RPM peak is, in fact, a true resonance. However, the 1200 RPM amplitude peak is not resonance due to the lack of a corresponding phase shift. The question now is; "What would cause a peak in 1X RPM vibration which is not due to resonance?"

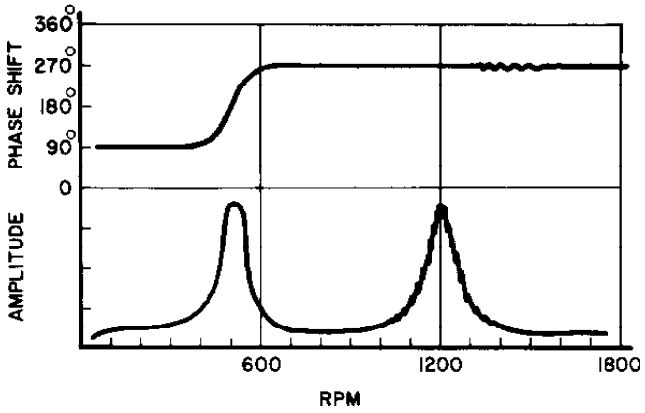

FIGURE 13 (Ref. 10)

One possible cause of the 1200 RPM amplitude peak in Figure 13 might be the presence of significant background vibration at the indicated frequency. For example, suppose a plot were being taken during the coastdown of an 1800 RPM motor, and at the same time, a 1200 RPM fan operating nearby was contributing *substantial background vibration at 1200 CPM*. Obviously, as the 1800 RPM motor coasts down, its speed must eventually coast through 1200 RPM. When this happens, the motor vibration will momentarily add to the background vibration from the fan to produce an amplitude peak at 1200 RPM.

EXAMPLE #2:

Referring to the amplitude and phase plots in Figure 14, an amplitude peak is noted at a speed of approximately 600 RPM.

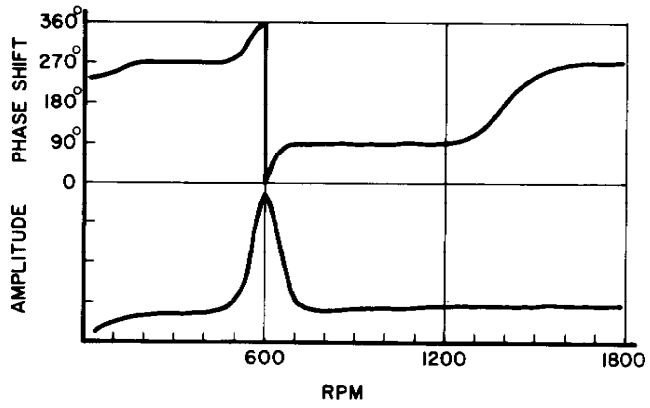

FIGURE 14 (Ref. 10)

A 180° phase shift corresponding to the amplitude peak confirms this as being a resonance. In addition, there is also a 180° phase shift at approximately 1400 RPM - but there is no amplitude peak at this speed to suggest resonance. The 180° phase shift, however, verifies that there is, in fact, a resonance at 1400 RPM. The question here is; "How can there be a resonance without a peak in amplitude?" The absence of the amplitude peak at resonance might be the result of one of the following:

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a) If the exciting force corresponding to the resonant frequency is very low, or if the system is *heavily damped*, little amplification in amplitude may occur at resonance. (See Figure 15 for the effect of damping on system response).

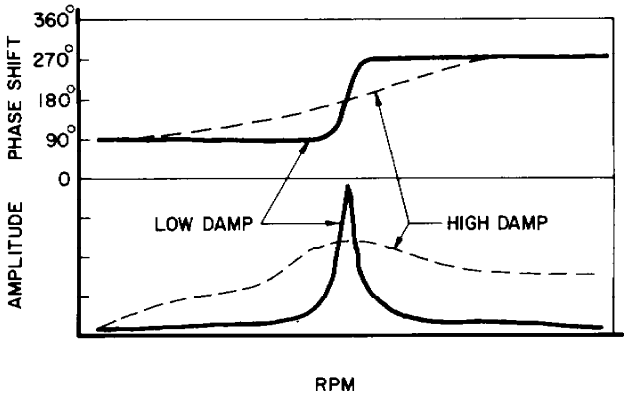

FIGURE 15
HIGHLY DAMPED VERSUS A LOWLY DAMPED RESONANT RESPONSE (Ref. 10)

b) *If the transducer is located at a "nodal point"* of the resonant system, little or no noticeable increase in amplitude may be noted at the resonance frequency. When a rotor shaft or structure is excited to vibrate at resonance, it will likely assume one of the vibratory modes illustrated in Figure 16. Each mode has one or more nodal points which are points of minimal amplitude. Of course, if the transducer is located at a nodal point, there may be little noticeable amplification of the vibration amplitude when passing through this resonance. However, the phase plot will reveal the characteristic 180° shift. *To prevent this from causing one to miss potential natural frequencies, the structure should be impacted in several different locations as shown in Figure 17.*

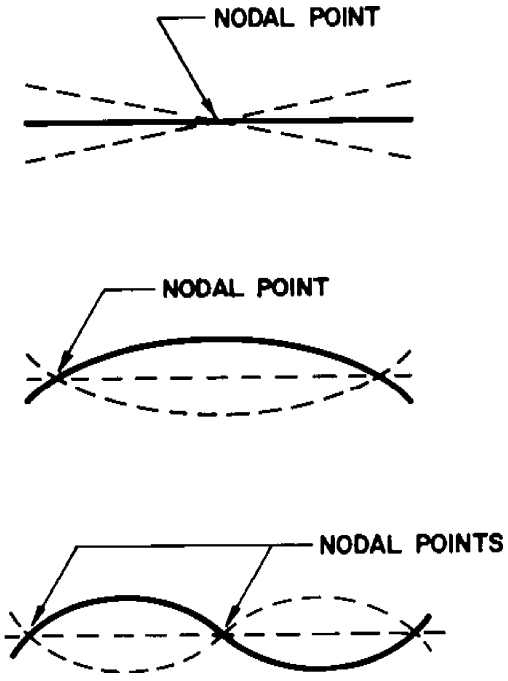

FIGURE 16
NODAL POINTS (Ref. 10)

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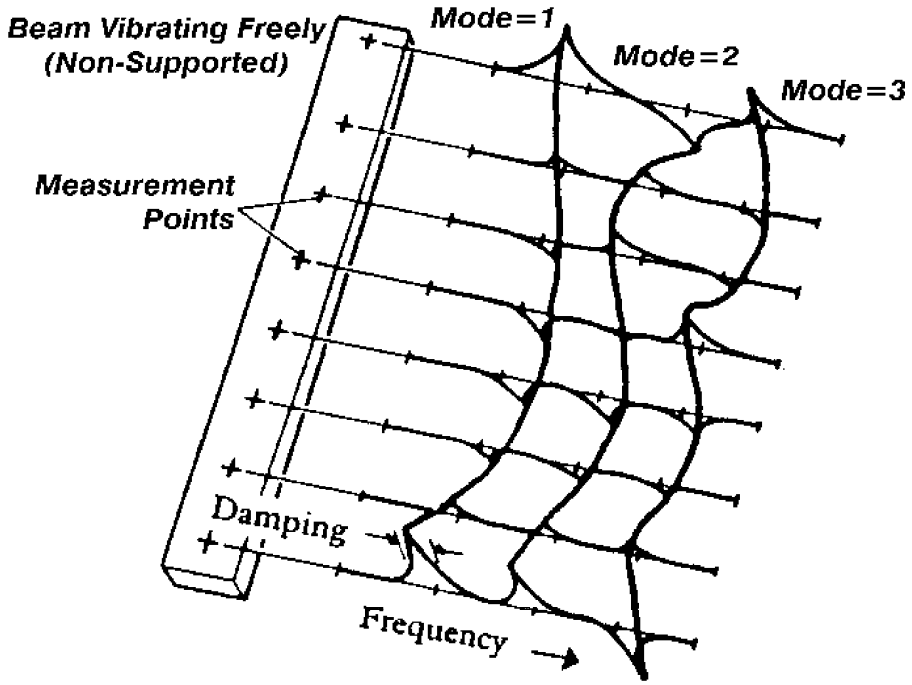

Damping, Frequency - Same at Each Measurement Point.

Mode Shape - Obtained at Same Frequency From all Measurement Points.

FIGURE 17
EXCITATION OF SEVERAL MODES AT VARIOUS POINTS OF A STRUCTURE (Ref. 4)

EXAMPLE #3:

Referring to the amplitude and phase-versus-RPM data in Figure 18, it is noted that an amplitude peak occurred at approximately 900 RPM. However, checking the phase information it is noted that the amplitude peak is accompanied by a phase shift of approximately 360° instead of the usual 180°. In this case, the phase data suggests that *there are actually two systems in resonance at or near the same frequency*, and each of the two resonances is contributing a 180° phase shift for a total phase shift of 360° *(these are called closely coupled modes)*. In this case identifying and correcting only *one* of the resonant systems may not totally solve the problem. *One must use either more lines of resolution and/ or a smaller frequency span to separate these 2 natural frequencies.*

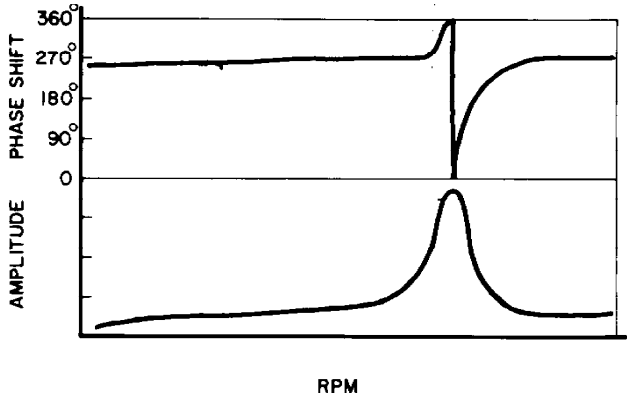

FIGURE 18 (Ref. 10)

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EXAMPLE #4:

Referring to the sample data in Figure 19, the somewhat unusual portion of the data which deserves explanation is the "dip" in vibration amplitude which occurs at approximately 1000 RPM; along with the accompanying 360° shift in phase. As in EXAMPLE #3, the 360° phase shift at 1000 RPM in Figure 19 *suggests two spring-mass systems in resonance at or very near the same frequency.* The dip or reduction in vibration amplitude at this frequency, sometimes referred to as a *"anti-node"* can be explained in Figure 20.

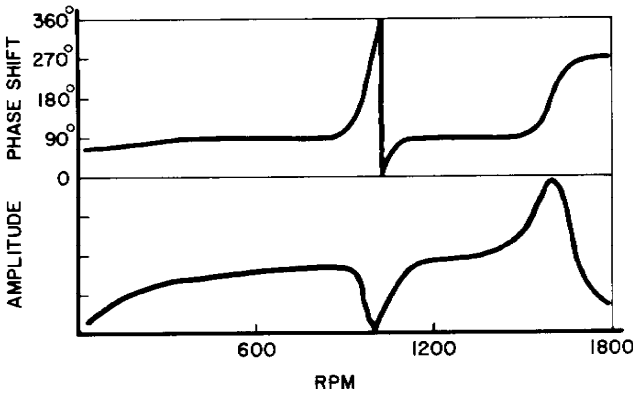

FIGURE 19 (Ref. 10)

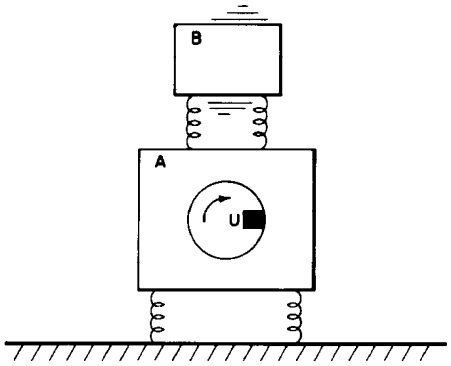

FIGURE 20
DYNAMIC VIBRATION ABSORBER (Ref. 10)

First, assume that the initial exciting force is the unbalance, "U". If spring-mass system "A" is resonant with the unbalance frequency, then we know from our previous discussions that the *actual vibration* of mass "A" lags the unbalance force by 90°. Now, if spring-mass system "B" is resonant at the same frequency, then its exciting force will be derived from the motion of mass "A". Therefore, the actual vibration of mass "B" must lag the motion of "A" by 90°. As a result, the actual vibration of mass "B" will lag the initial unbalance force by a total of 180° (90° + 90° = 180°). This produces two opposite forces acting on mass "A", one being the initial unbalance force and the second being the force of vibration of mass "B". Since the two forces acting on mass "A" are opposite, mass "A" will exhibit a minimal vibration amplitude as illustrated by the dip in Figure 19. Mass "B", on the other hand, may reveal an extremely high amplitude of vibration. For this reason, when the data suggests a possible anti-node, it would be advisable to identify the second system in resonance and plot its response as well, to insure satisfactory performance. *Figures 19 and 20 illustrate how and why a dynamic vibration absorber is often very effective.*

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EXAMPLE #5:

Plots similar to that in Figure 21 are sometimes obtained when non-contact pickups are used to identify rotor critical speeds. The plot does display a peak amplitude and the normal 180° phase shift. However, it should be further noted that the amplitude plot includes a "dip" at approximately 5000 RPM, and, in this case, just slightly below the shaft critical speed. The amplitude dip in Figure 21 does not have a substantial corresponding phase shift and, thus, is not likely the result of an anti-node, plotted in Figure 19. The cause of the indicated amplitude reduction (dip) in the plot, Figure 21 can usually be traced to *excessive electrical and/or mechanical run-out of the shaft at the non-contact pickup target area*. The non-contact or proximity pickup cannot distinguish between actual shaft vibration and any run-out or eccentricity of the shaft journal. As a result, the non-contact pickup provides a signal proportional to the *vector sum* of run-out and actual shaft vibration.

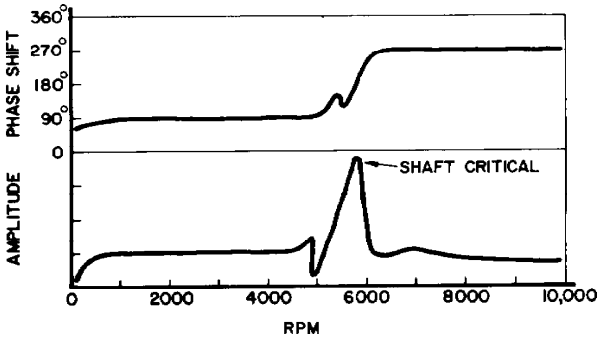

FIGURE 21 (Ref. 10)

Operating well below its critical speed, a shaft is considered to be rigid and, thus shaft vibration will essentially be in-phase with unbalance. However, as the RPM of the shaft approaches its critical speed, shaft deflection will progressively increase. In addition, the deflection in the shaft (actual shaft vibration) begins to lag the heavy spot of unbalance. In fact, operating at critical speed, shaft deflection lags the unbalance heavy spot by 90°. and at operating speeds above shaft critical, deflection will lag the unbalance by 180°. The important point to note here is that the amplitude and phase of shaft vibration is undergoing a change as the shaft passes through critical speed. As a result, the vector sum of shaft vibration and run-out is also undergoing a change. The dip or amplitude reduction plotted in Figure 21 results when the shaft vibration and run-out vectors become momentarily out-of-phase and, thus, tend to cancel one another. Of course, instead of canceling one another, it may occur that the run-out and shaft vibration vectors will momentarily add to reveal a plot such as that in Figure 22. In either case, it is important to note that when using non-contact pickups, the presence of excessive run-out may give a very distorted picture of rotor response. The recorded data may suggest a condition which is far better or far worse than the true response of the rotor. Therefore, it is advisable to carefully measure run-out amplitude before response measurements are taken. Of course, *excessive run-out should be physically or electronically eliminated so that plots of true rotor response can be obtained.*

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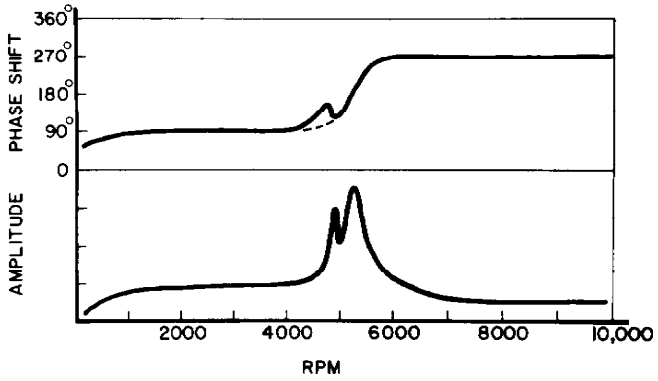

FIGURE 22 (Ref. 10)

EXAMPLE #6:

The recorded plot in Figure 23 illustrates a situation where the recorded data can be somewhat misleading. The suspicious data in the plot, Figure 23, are the somewhat unusual phase shifts that occur at 6000 and 7000 RPM. While these apparent phase shifts may suggest resonant frequencies which were not indicated on the amplitude plot, it is quite possible that these are not phase shifts at all. From our previous discussion you will recall that *two* signals are required to obtain a DC voltage proportional to phase for plotting purposes. These signals are 1) a voltage pulse from a reference pickup at the shaft of the machine and, 2) a vibration signal from a vibration pickup. If either one of these signals is lost or lacks sufficient amplitude, the phase voltage will drop to zero. Referring to the amplitude plot in Figure 23, you will note that the vibration amplitude is at a relatively low value between 6000 and 7000 RPM. Note also that the phase plot drops to 0° between 6000 and 7000 RPM. Therefore, *it is more likely that the apparent phase shifts are nothing more than the absence of a sufficient vibration signal to provide a true phase reading*. Below 6000 and above 7000 RPM the vibration amplitude has increased sufficiently to provide phase indications.

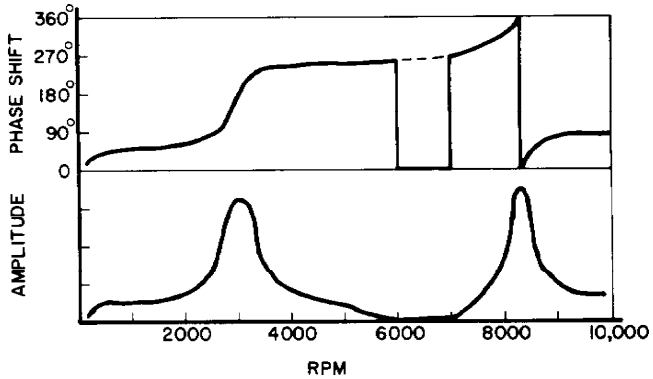

FIGURE 23 (Ref. 10)

Bode' Plots are not without limitations. While they readily reveal natural frequencies of the machine or structure or the entire spring-mass system, Bode' plots alone may not reveal the response of the system to other vibration frequencies which can be affecting the overall performance of the machine. The analyst needs to be aware of this. Remember, they are taken from filtered data which shows the response of this system as the speed (or frequency) is brought down or up through these frequency regions.

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4.42 POLAR PLOTS

In the previous section, Bode' Plots were used to identify natural frequencies or resonances. Recall that Bode' Plots were amplitude-versus-RPM and phase-versus-RPM plots. These were recorded on a Cartesian (rectangular) coordinate graph.

Another technique for presenting this information is called the "polar" plot. Instead of using a rectangular graph, polar plots are displayed on a polar coordinate graph. The polar plot is a *presentation of amplitude and phase only*, unlike Bode' plots.

4.421 Setting Up for Polar Plots (Ref. 10)

As in the case of Bode' plots, the instrument used needs to be equipped with a tracking filter. It also needs polar amplitude and phase output signals to drive a standard X-Y recorder.

The pen on the X-Y recorder needs *to start at the center point* of the polar graph. To do this the X-Y recorder is adjusted so that a zero-amplitude signal will position the pen at the center of the graph. The *FULL SCALE amplitude control* on the recorder is adjusted so that as the vibration amplitude increases it *will cause the pen to move radially outward from the center*. The amount that the pen moves away from the center is proportional to the amount the vibration increases. The phase of the vibration will determine the angular direction in which the pen will move as the vibration increases.

4.422 Advantages of Polar Plots Over Bode' Plots (Ref. 10)

1. Provides *an immediate indication of the unbalance vector* without the need to compare separate amplitude and phase plots.

2. Eliminates confusing phase discontinuities that appear on Bode' plots as the phase changes from 0° to 360° (which occurred at the 2400 CPM natural frequency on Figure 12).

3. Eliminates phase discontinuities that appear on Bode' plots when the vibration amplitude has reduced to a low level where there is not sufficient vibration signal to maintain a phase indication. See Figure 23.

4. Where only a single-pen recorder is available, the Polar plot allows both amplitude and phase data to be obtained during a single run-up or coastdown of the machine, since a Polar plot only requires a single pen anyway.

5. Where non-contact pickups are being used, the vector resulting from combined mechanical and electrical run-out can be eliminated by recalibrating the recorder amplitude ZERO control to the origin of the polar graph while the run-out vector is being displayed with the machine on slow-roll. Figure 24 shows comparative Polar plots obtained with and without recalibrating for shaft run-out. As illustrated, run-out can result in substantial unbalance vector at any one operating speed.

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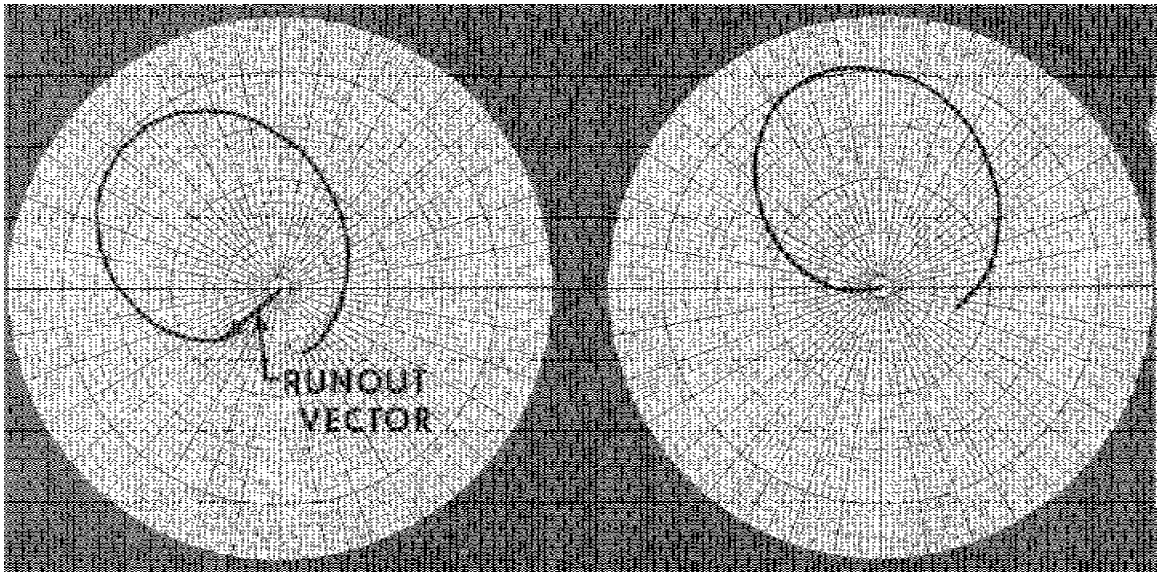

FIGURE 24
POLAR PLOTS WITH AND WITHOUT RUNOUT COMPENSATED (Ref. 10)

4.423 Comparison of Bode' and Polar Plots for Natural Frequency Testing

The plots in Figure 25 illustrate how a polar plot compares with a familiar Bode' plot. Note that in the Bode' plot, there are two distinct peaks in the lower, amplitude-versus-RPM plot. One is at approximately 2400 RPM and the other is at approximately 4900 RPM. Are these peaks resonant?

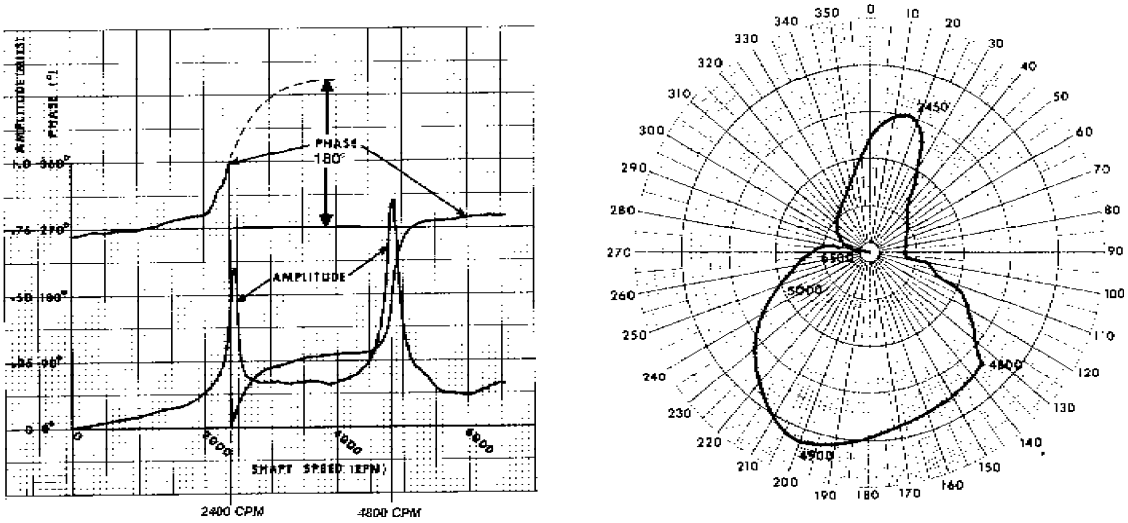

FIGURE 25
COMPARING POLAR AND BODE' PLOTS (Ref. 10)

By looking at the 4900 CPM peak, it is easy to conclude that it is resonant because of the corresponding 90° phase shift in the upper phase-versus-RPM plot (then 180° phase change when it passed through resonant amplification). But is the 2400 CPM peak resonant? If the phase trace at the base of the 2400 CPM peak is transposed to the trace at 360° as shown by the dashed lines in the figure, the 180° phase shift can be seen here also. It is resonant.

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As a comparison, notice that orbits in the polar plot are 180° apart (at the unbalance vectors). The resonance response is clearly displayed, although the RPM's had to be manually entered.

4.424 Applying Polar Plots to Natural Frequency and Resonance Diagnostics (Ref. 10)

While polar plots can be used to diagnose problems not caused by or connected with resonance, this discussion will focus on using polar plots in resonance diagnostics only. Polar plots are obtained at each bearing of the machine. Usually two non-contacting pickups are used, placed 90° apart usually in the 9:00 and 12:00 positions, but preferably in the 10:30 and 1:30 positions to be removed from the split in the bearing cup halves (45° off vertical at each location). The polar plots from each bearing are compared to make a final analysis. They will provide "mode shape" information of each resonance, if it is present.

To illustrate, assume that the two plots in Figure 26 were obtained at the two bearings of a large turbine during start-up. Note that both plots reveal amplitude peaks at speeds of *1200, 2200 and 3000 RPM*. *The amplitude peaks together with the corresponding shifts in phase confirm that all three are resonant frequencies.* In addition, it can be noted that both bearings show the same phase indication for the resonant amplitude peak at 1200 RPM. *This suggests that the resonance at 1200 RPM is the first rigid mode of the bearings (structure) diagrammed in Figure 27*. Rigid modes are associated with structured support natural frequencies which typically occur below the first bending mode of the rotor. Thus, the rigid modes of structure support have lower frequencies than those of the bending modes of the rotor. Further, the plots show that the amplitude peaks at 2200 RPM have opposite (180°) phase. This would indicate that the resonance at 2200 RPM is the *second rigid mode of the bearings* as illustrated in Figure 27. Finally, since the amplitude peaks at *3000 RPM again show* identical phase indications on the two plots, it can be concluded that the resonance at *3000 RPM is the first bending critical of the turbine rotor as illustrated in Figure 27.* Although this same information and interpretation could be obtained from Bode' plots, *the comparison is made considerably easier with polar amplitude and phase plots.*

4.425 Limitations of Polar Plots

Despite advantages and many useful applications, Polar plots also have limitations when compared to amplitude and phase-versus-RPM plots. Major limitations include:

1. While the *Polar plot does* reveal the presence and significance of resonant conditions during run-up or coastdown, it *does not provide a speed reference to indicate the RPM at which resonance occurs*. As a result, it is necessary to monitor machine speed and manually record RPM values as noted on the sample plots in Figures 25 and 26.

2. *The Polar plot is a plot of the unbalance (synchronous) vibration and, thus, is restricted to examining only the vibration occurring at 1X RPM*. As indicated previously, severe resonant conditions can be excited by vibration frequencies other than 1X RPM unbalance. Or, during start-up or coastdown, additional problems such as oil whirl, resonance whirl or rubs may occur resulting in nonsynchronous vibration frequencies. *Evidence of these and other problems can be seen when the overall or unfiltered vibration is plotted versus RPM. Such problems are likely to go undetected if only Polar plots are obtained.*

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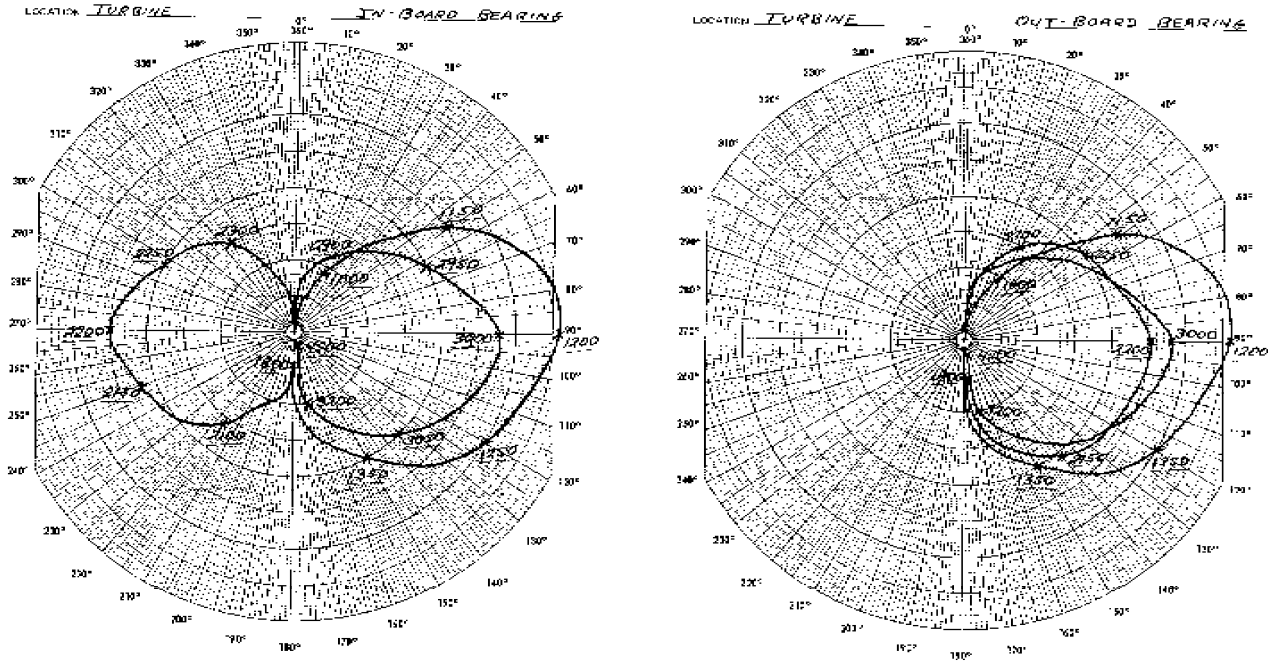

FIGURE 26
COMPARATIVE POLAR PLOTS OBTAINED FOR EACH BEARING OF THE MACHINE CAN AID IN IDENTIFYING RESONANT MODE SHAPES (Ref. 10)

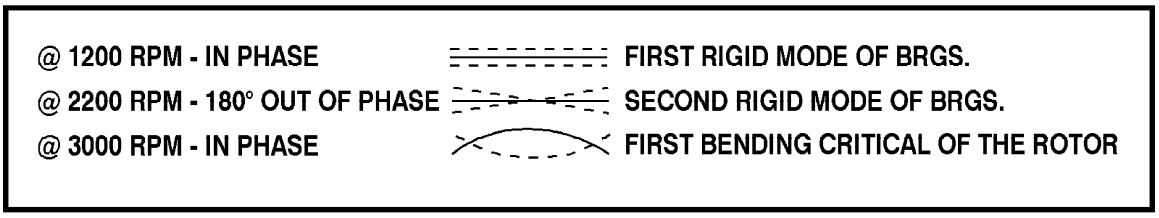
@ 1200 RPM - IN PHASE
FIRST RIGID MODE OF BRGS.
@ 2200 RPM - 180° OUT OF PHASE
SECOND RIGID MODE OF BRGS.
@ 3000 RPM - IN PHASE
FIRST BENDING CRITICAL OF THE ROTOR

FIGURE 27
RESONANT MODE SHAPES

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CHAPTER 5

ENHANCED VIBRATION DIAGNOSTICS USING CASCADE DIAGRAMS

5.0 INTRODUCTION

Cascade diagrams (sometimes referred to as "Waterfall Plots" or "Spectral Maps"), give the analyst a view of the machine vibration characteristics relative to operating speed or the actual running time. *The cascade plot graphically displays resonant events, rotor rubs, oil whirl, oil whip, beat frequency events and other mechanical faults in a series of frequency spectra which are generated as the machine changes speed, or as time progresses*. Signals that are changing slowly with respect to time are difficult to see in real time. However, periodic sampling with waterfall plots can clearly show these changes which might otherwise have gone unnoticed.

For example, in the synthetic fibers industry which employs many winders (for winding small strands of yarn filament onto bobbins up to a package size of approximately 25 to 30 lbs), these machines often require anywhere from 30 or 40 minutes up to about 8 or 9 hours to wind a full package (see drawing of such a winder in Figure 8), these machines are forced to pass through several resonances which can often be quite severe. By examining waterfall plots captured throughout the package build, one can often see when the machine passes through resonant events, or when it encounters beat frequencies increasing amplitude for a few minutes (when either the speed or a running speed harmonic of one component closely approaches the operating speed of another component). If this entire package build (known as a "doff cycle") is captured on tape, or within the memory buffer of a spectrum analyzer, this will allow one to closely examine these resonant and beat frequency events, even to the point of calculating the resonant amplitude amplification factor (Q), as well as the damping factor (ζ). This will be covered in later portions of this chapter and portions of other chapters in the text.

5.1 DIAGNOSIS OF ROTOR RUB PROBLEMS

Cascade plots are especially helpful in diagnosing radial rubs. An example cascade plot is shown in Figure 1 which illustrates *a partial rub*. This cascade plot shows how vibration amplitude and frequency changes with machine RPM. It shows a series of single spectra in an amplitude versus frequency plot captured as the machine is brought up to speed. Note that the 1X RPM peak goes through a natural frequency at about 1500 to 1800 RPM. Later, the amplitude at ½X RPM goes through the same natural frequency, at which point the amplitude of the ½X RPM peak is much greater than the peak at 1X RPM when the speed of the machine is approximately 3000 to 3600 RPM. *An unusual occurrence like this often indicates a partial radial rub. In these cases, one will have to ensure the subsynchronous frequency is .50X RPM (and not 40% to 48% of RPM which would point to another problem source - oil whirl)*.

Shaft orbit displays (known as "Lissajous Orbits") are also shown in Figure 1 and are likewise very helpful in diagnosing rubs. These orbit displays show the actual path the shaft itself follows inside the bearing (using the output from two eddy current proximity probes placed in the machine housing about 40 to 60 mils away from the shaft target, and with the probes located 90° apart from one another). The shape of the orbit is helpful in determining phase, as well as in distinguishing the nature of the rub, its severity, and probable frequency content. Lissajous orbits will be discussed separately in later sections of the seminar text.

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Partial rubs will generate noticeable subsynchronous vibration which is clearly captured within a cascade diagram, but very likely will be missed if just capturing a "Peak Hold Spectrum", or just observing the display of a spectrum analyzer in real time. ***For this reason, cascade diagrams should always be captured on critical machinery when conducting acceptance tests on new or rebuilt machinery***. If the subsynchronous vibration is caused by a partial rub, it will normally show integer subharmonics (1/2X or 1/3X or 1/4X RPM, etc.). *Several different integer subharmonics can simultaneously occur when a light rub with a lower radial force occurs*. Such a response is shown in Figure 2 which features both the spectral and the orbit responses for 1X, 1/2X, 1/3X, 1/4X, and 1/5X RPM. In this case, note that maximum vibration occurred at 1X RPM and each of its 5 subharmonics when they accelerated and passed through the fundamental bending mode (first critical speed) in the vicinity of 1700 to 2100 CPM (28.3 - 35.0 Hz).

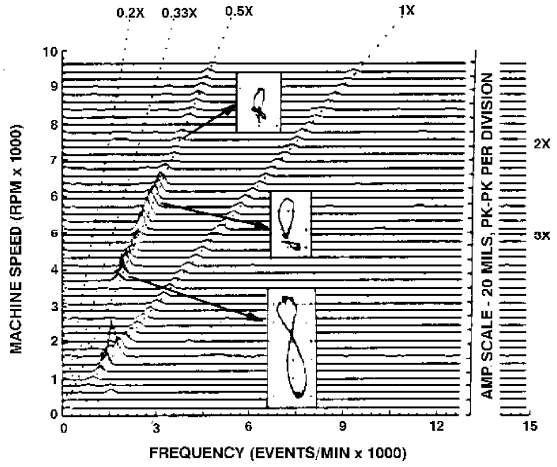

FIGURE 1
CASCADE DIAGRAM INDICATING A PARTIAL RUB EVENT UNDER HIGH FORCE (Ref. 3)

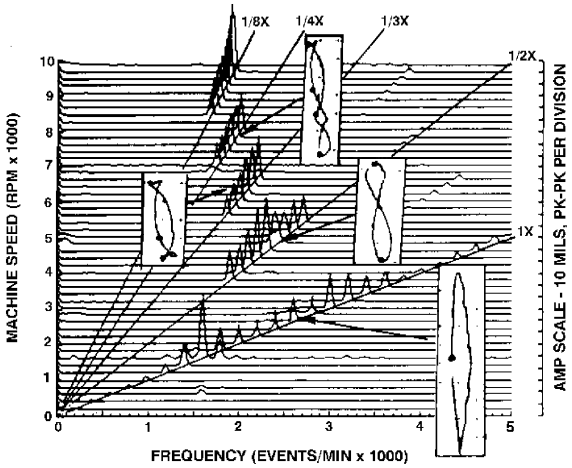

FIGURE 2
PARTIAL RUB EVENT WITH LOW RUB FORCE GENERATING MANY INTEGER SUBHARMONICS (Ref. 3)

Partial rub events can eventually lengthen in duration and impact finally leading to a full annular rub in which case a virtually continuous rub occurs. *Figure 3 shows a cascade diagram taken during a full annular rub of a seal rubbing on a shaft. The shaft orbit displays are also included*. Note the very "messy" orbit shown on the lower right of the plot which showed very unstable shaft motion within the bearing and seal. This plot shows how the response changed as the machine

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was brought up to speed. Note that at lower speeds, little vibration was noticed. At higher speeds approaching 3500 RPM, a response at about 7000 CPM began (at that instant when 7000 CPM equalled 2X RPM). It remained at this same 7000 CPM frequency, even when the machine was brought all the way up to 4500 RPM. *This 7000 CPM response corresponded to the rotor/seal system's natural frequency which was higher than the natural frequency of the rotor alone before it contacted the seal due to the additional stiffness supplied by the seal during the rub event*. This 7000 CPM frequency, then, was the "speed" of the backward precession (known as "reverse precession"). This phenomenon will be discussed in the chapter covering the **"Vibration Diagnostic Chart"** under the **"Rotor Rub"** section.

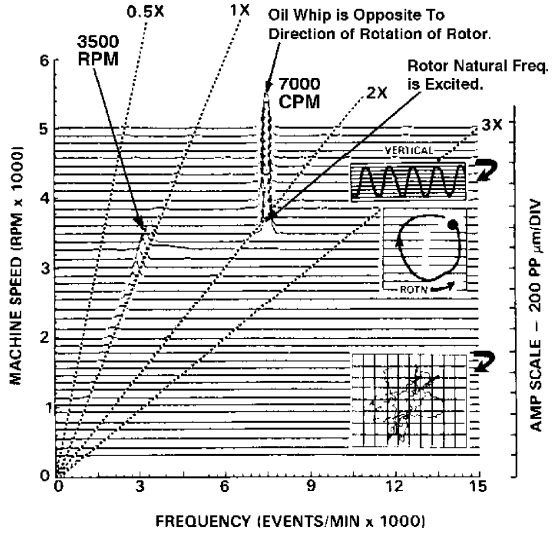

FIGURE 3
MULTIPLE IMPACTING SEAL RUB WHICH GENERATED REVERSE PRECESSION AT APPROXIMATELY 3500 RPM EXCITING A 7000 CPM ROTOR/SEAL SYSTEM NATURAL FREQUENCY (Ref. 3)

Figure 4 shows another type of full annular rub which is dominated by the response at 1X RPM followed by lower amplitude harmonics of running speed. In this case, the full annular rub was dominated by forward precession synchronous speed, and began experiencing subsynchronous vibration when the rotor accelerated to approximately 9000 RPM.

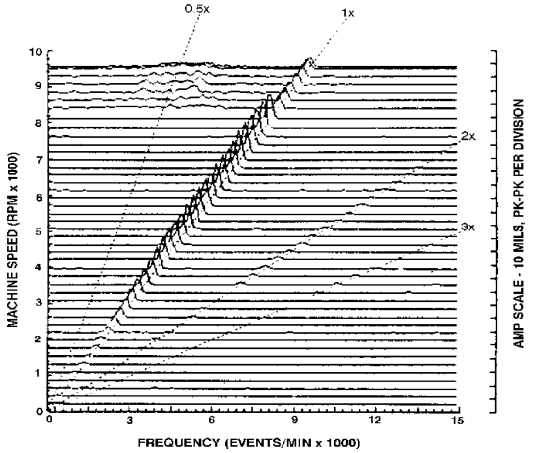

FIGURE 4
FULL ANNULAR RUB DOMINATED BY FORWARD PRECESSION SYNCHRONOUS SPEED (Ref. 2)

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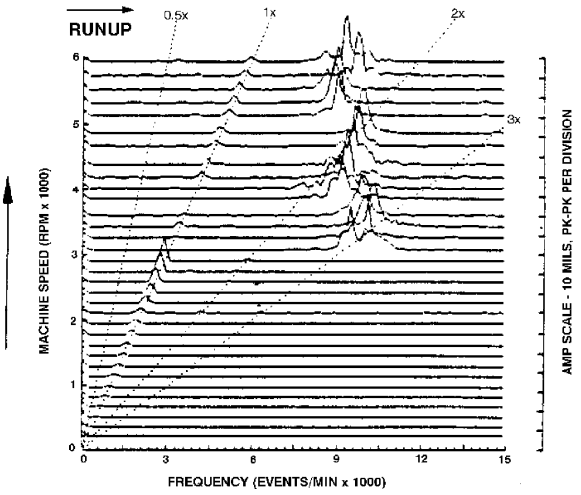

FIGURE 5
FULL ANNULAR RUB DURING RUNUP WITH SELF-EXCITED BACKWARD PRECESSION (Ref. 2)

Figure 5 is an example of a full annular rub where a self-excited backward precession has occurred at a speed just below the first critical of the rotor. Note that even though the operating speed frequency (1X RPM) kept increasing, a wideband response of energy remained in the vicinity of 9000 CPM - *a clear indication of a potentially severe rub event. However, Figure 5 clearly shows this could only be seen by examining a cascade diagram within a spectrum analyzer* (or alternatively sent to a plotter from analyzer memory).

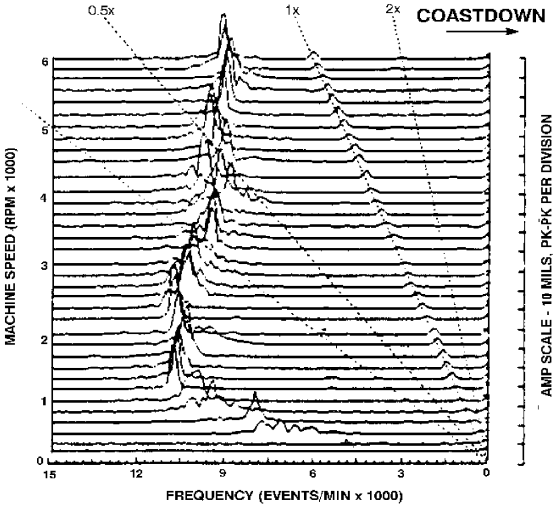

FIGURE 6
FULL ANNULAR RUB DURING COASTDOWN
(NOTE CONTINUED BACKWARDS PRECESSION UNTIL VERY LOW SPEED)
(Ref. 2)

Interestingly, note that the backward precession remained until the same rotor nearly came to a stop even after the unit was shut off and coasted down as shown by Figure 6. *Once again, this was only clearly observed with a cascade diagram*.

Note that, in both cases, when the backward precession was occurring, the comparative amplitude at 1X RPM was very low. Since the friction remained throughout the rub, it introduces non-linearities that generate many harmonics of running speed as well.

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5.2 DIAGNOSTICS OF SERIOUS OIL WHIRL AND OIL WHIP PROBLEMS

Very large turbomachinery and compressors outfitted with journal bearings are usually outfitted with proximity probes set 90° apart, usually 45° to the left and right of vertical (10:30 and 1:30 o'clock positions). In this way, these critical machines can be permanently monitored. Also, this will allow an analyst to capture both cascade diagrams and shaft orbit displays similar to that shown in Figure 7 during startups and coastdowns.

On machines that are equipped with hydrodynamic journal bearings, the most likely cause of subsynchronous instability is oil whirl (which will be discussed in the chapter covering the **"Vibration Diagnostic Chart"**). Figure 7 shows the development of oil whirl just after the shaft was brought up to speed. *Note that the shaft went into oil whirl at about 1800 RPM and remained in whirl until approximately 4000 RPM*. At this point, note that the shaft 1X RPM began entering resonance which caused enough force to actually suppress and overcome oil whirl (*during oil whirl, the highly pressurized oil wedge actually drives the shaft around rather than the shaft itself driving the oil wedge as the shaft should*). However, once the machine had passed through resonance, the shaft forces and resultant vibrations drop off substantially allowing the oil whirl frequency to reappear as shown in Figure 7 which is inherently unstable.

The effect from oil whirl disappears because there is enough force from the natural frequency to start moving the oil wedge again.

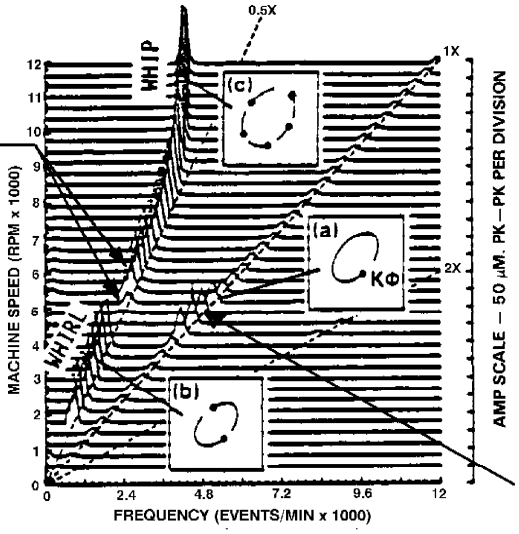

FIGURE 7
DEVELOPMENT OF OIL WHIRL JUST AFTER STARTUP
(FOLLOWED BY OIL WHIP FROM 9200-12,000 RPM) (Ref. 3)

When the rotor speed increased to just above 9200 RPM, it equalled about 2.2X its critical speed. At this time, the oil whirl which was approximately 45% of RPM was brought into coincidence with this critical speed. *At this point, the oil whirl was suddenly replaced by oil whip*. *The oil whip frequency remained at the same 9200 CPM, independent of rotor RPM, even though the machine continued up to 12,000 RPM. In other words, the oil whirl frequency "locked into" the rotor natural frequency, thereby becoming oil whip* (a highly unstable rotor problem). When a shaft goes into oil whip, its controlling dynamic factors become mass and stiffness in particular, with its resulting amplitude limited only by the bearing clearance. Note the high amplitudes at the 9200 CPM oil whip frequency in comparison to running speed (1X) in Figure 7. *Left undetected and uncorrected, this serious instability can lead to catastrophic failure costing millions of dollars on these machines (not only in parts and manpower, but also in downtime). Very likely, only the use of cascade diagram capture could have detected (and properly diagnosed) this severe problem!*

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5.3 DIAGNOSTICS OF RESONANT FREQUENCIES

Cascade diagram acquisition is a powerful technique that enables an analyst to see what exactly is happening to his machine as it accelerates (or decelerates) from rest to operating speed. *Such a plot can point directly to potential natural frequencies which must be avoided*.

EXAMPLE A: DETECTION OF SYNTHETIC FIBER TEXTURING WINDER RESONANCE DURING OPERATION BY CAPTURING OF CASCADE DIAGRAMS

Figure 8 is a sketch of texturizing winder showing the vibration measurement points for the test and also the location of the major components of the winder. This constant surface-speed machine uses a friction roll (drive roll) to drive each of two bobbins held by what is called a chuck shaft (see Figure 8). When fully wound with yarn filament, each bobbin package normally weighs about 25 to 30 pounds.

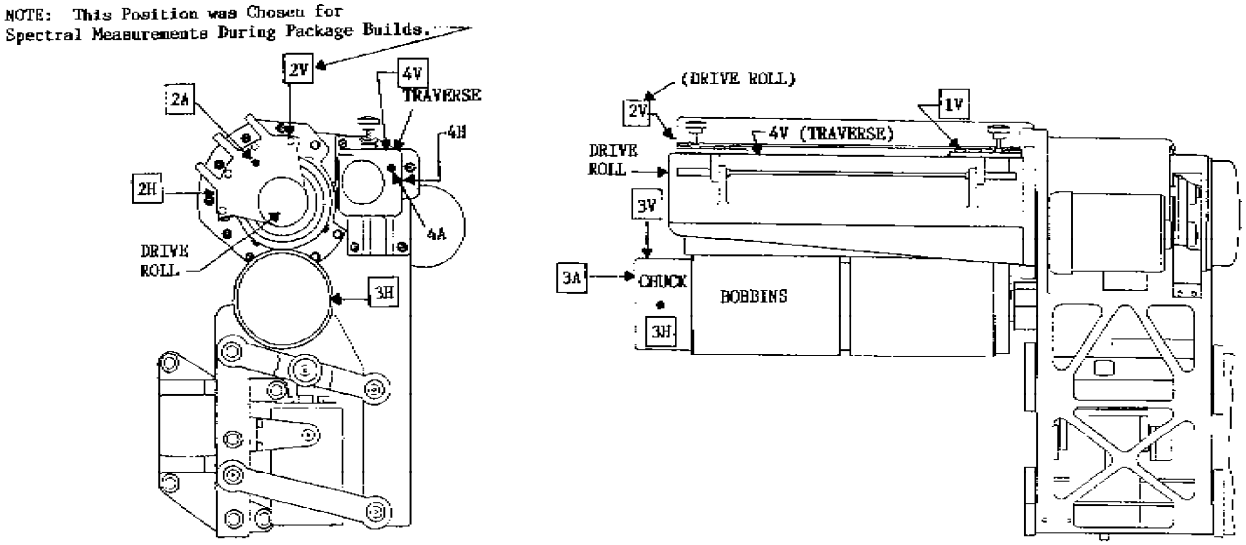

FIGURE 8
MEASUREMENT LOCATIONS FOR SYNTHETIC FIBERS WINDER

A controlled coastdown is shown in the cascade diagram of Figure 9 captured as the winder built a full 30 lb package on each bobbin. The friction roll remained at about 2670 RPM throughout the test. However, the chuck outfitted with the bobbins continually decreased in speed from 1620 to 855 RPM even though its surface speed remained constant (ft/min). Note the constant 2670 RPM friction roll speed in the diagram. But also note the first, second and third harmonic chuck speeds which all decelerated with increasing package size. Over on the right hand side are shown the chuck RPM, along with the vibration at 1X and 2X chuck RPM. Note the increase in amplitude as 2X chuck speed coincided with the friction roll speed of 2670 RPM at Point "A" on the diagram (chuck speed was 1330 - 1370 RPM in this region). The resultant *beat frequency* between 1X friction roll speed and 2X chuck speed caused an increase from .109 up to .372 in/sec, which was not unexpected.

After passing through this area, 2X chuck speed peak dropped back to .127 in/sec as shown in the plot. After this, vibration at 2X chuck speed was relatively low throughout the remainder of the package build. However, vibration at 1X chuck speed began increasing as the chuck speed decreased to 1035 RPM and peaked out at 885 RPM with an excessive amplitude of *1.79 in/sec* as shown in the plot (and in the annotation just to the right of the plot). After passing through the apparent resonance, the amplitude at 1X chuck RPM

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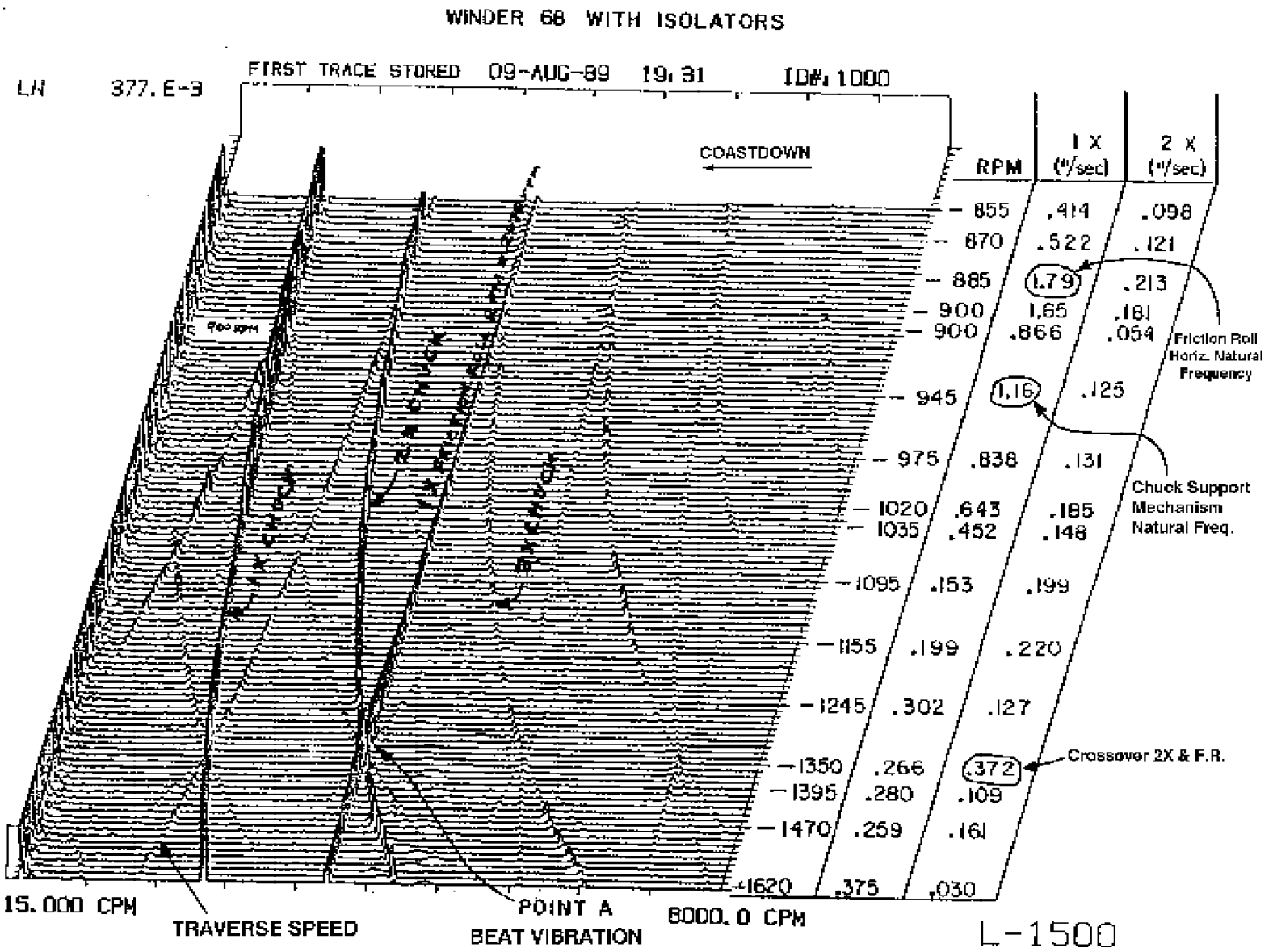

FIGURE 9
CASCADE DIAGRAM CAPTURED DURING A CONTROLLED COASTDOWN ON THE WINDER

decreased to .414 in/sec at 855 RPM as shown. After generating this cascade plot, impulse tests were performed on the chuck support mechanism and friction roll *which indicated natural frequencies at approximately 990 CPM in the horizontal direction and 1050 CPM in the vertical direction. Also, the friction roll had a horizontal natural frequency at about 900 CPM*. This indicates the reason for high vibration shown by 1X chuck RPM at speeds between 885 and 1020 RPM. *This severe problem was vividly displayed by use of cascade diagrams. Here, the solution was application of damping materials rather than altering the chuck and friction roll natural frequencies* since this was considered impractical and time consuming at this time for this variable speed machine (controlled by an inverter).

By referring to Figure 9, one can see the imminent coincidence of the natural frequency at about 900 CPM begin to occur when 1X chuck speed slowed to approximately 945 to 975 RPM. At other times, cascade plots taken during runups or coastdowns can reveal very strange behavior as shown below:

EXAMPLE B: DETECTION OF A GODET ROLL ASSEMBLY VIBRATION PROBLEMS FOR THIS SYNTHETIC FIBERS MACHINE USING CASCADE DIAGRAMS

Figure 10 is a sketch of a Godet Roll Assembly for a synthetic fiber operation which shows the various vibration measurement points as well as the identification of the rolls. *Rolls 3A and 3B* (each of which were heated) *were supposed to be capable of operating up to 6000 RPM at a production rate of approximately 3000 meters/min* (MPM). *Unfortunately, when they tried to operate the panel at speeds just above 4000 RPM (2000 mpm), they encountered high vibration problems which resulted in product quality problems*. The client desired Technical Associates to (1) determine the cause of such high vibration at operating speeds far below that rated for the machine; and (2) submit recommendations which would resolve this problem and truly allow them to operate the machine up to 6000 RPM.

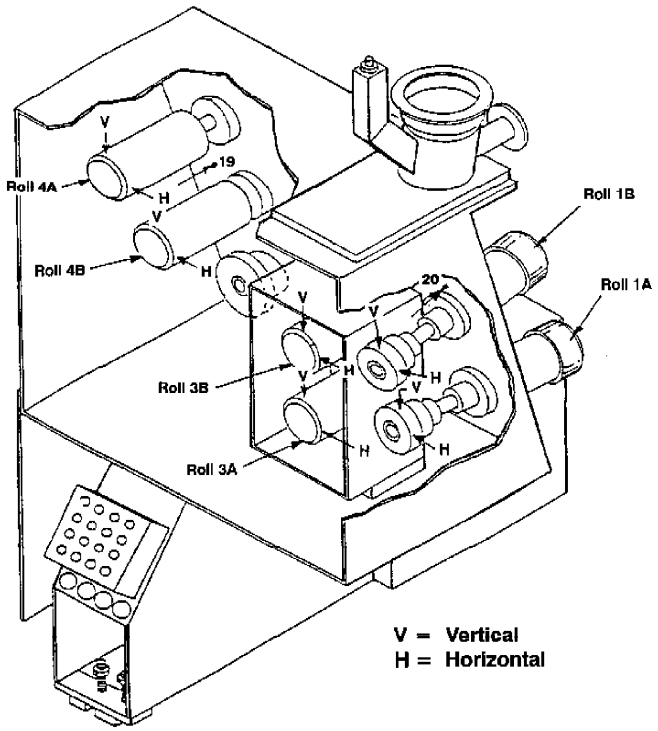

FIGURE 10
GODET ROLL PANEL NATURAL FREQUENCY MEASUREMENT POSITIONS

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Please refer to the cascade diagram of Figure 11 which shows results of a controlled runup test for Godet Roll 3A using an AC motor controlled by an inverter. Several interesting things appear on the waterfall of Figure 11. First, *one can see the resonances at about 2730 CPM, 4250 CPM, 5978 CPM and then in an entire region from approximately 6400 to 7100 CPM. This plot also shows something else very unusual which began to occur at a speed of approximately 6300 RPM. Note the subsynchronous frequencies* that began to appear at this point and remained up to a speed of approximately 6800 RPM. *While one set of subsynchronous frequencies was increasing in frequency with an increase in speed, another subsynchronous set was actually decreasing with increasing rotor speed*. Later, this was determined to be an instability of the Godet Roll bearings. After much study, it was determined that the bearings had improper preload (exerted by wavy springs) and also improper interference fit. Therefore, when the rotor began going into the resonant amplification zone between 6400 and 7100 RPM, this *bearing instability* would occur.

Figure 12 shows the waterfall spectrum during a controlled runup test up to 10,800 RPM after a series of modifications were made to the Godet Roll. *Note the complete elimination of the structural resonance problems and also note that the roll fundamental natural frequency was successfully moved from about 6000 up to 8520 RPM*. In addition, even though the rotor had the 8520 RPM fundamental natural frequency, it now suffered much lower amplitudes of *only .260 in/sec* even when resonant compared to a level of *.953 in/sec at the rotor fundamental before.* This was due to a series of modifications including changes in shaft diameters, material changes and both damping and isolation treatments.

Interestingly, note that none of the bearing instability previously encountered at speeds of only 6100 RPM occurred again until a speed of about 10,800 RPM was attained. Strangely, at this point, a subsynchronous frequency at exactly 2X the bearing cage defect frequency would occur almost every time this was attempted (cage frequency was .411 times RPM). At this time, the predominant vibration would not be at 1X rotor RPM, but at the 2X bearing cage frequency. Reasons for instability at this particular frequency were never completely understood. *However, as long as roll speeds remained below 10,000 RPM, this bearing instability would not occur*.

The most important success with regard to the client was that they now had a machine which could successfully run not only up to the desired minimum of 6000 RPM, but now up to as high as 8000 RPM with amplitudes remaining below .200 in/sec. In addition, *the vendor now had a machine which could enter a whole new market up to speeds of 8000 RPM which corresponded to production speeds of about 4200 MPM*.

Note from the cascade diagrams in this section that during a runup test, all peaks which are harmonic or subharmonic orders of running speed closely approximate straight lines. As the machine increases in speed, the slope of the "lines" are skewed to the right, the slope depending on the running speed order (or harmonic). On the other hand, during a coastdown, the slope of the "lines" are skewed to the left as the machine coasts to a stop. In both cases, spectral peaks which are not harmonically related (nonsynchronous or asynchronous) or those that represent instability, move in random directions on the plot.

This section enabled the analyst to see that transient events, rubs, oil whirl, oil whip, resonance and bearing instability can actually be "seen" with the use of cascade plots. The behavioral patterns as well as frequencies could be identified as they interacted with others. *This is truly an important diagnostic "tool in the vibration analyst's tool box".*

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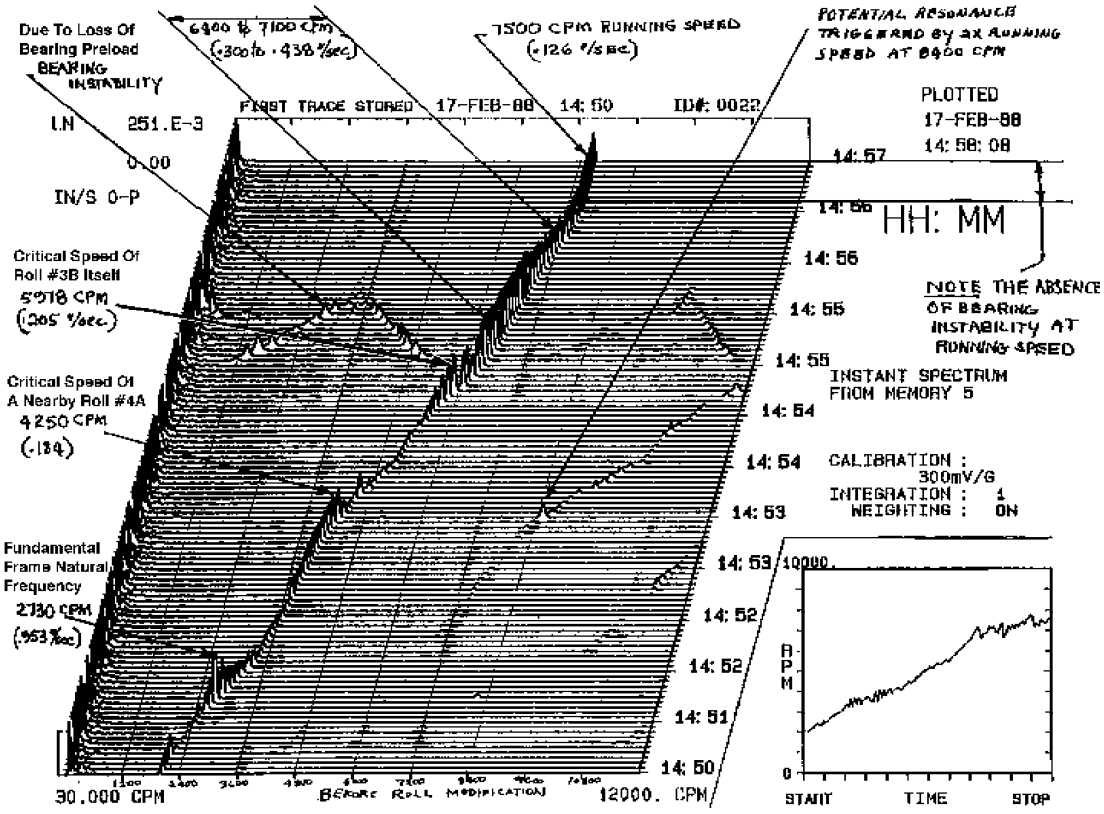

FIGURE 11
CONTROLLED RUNUP TEST FOR GODET ROLL 3A

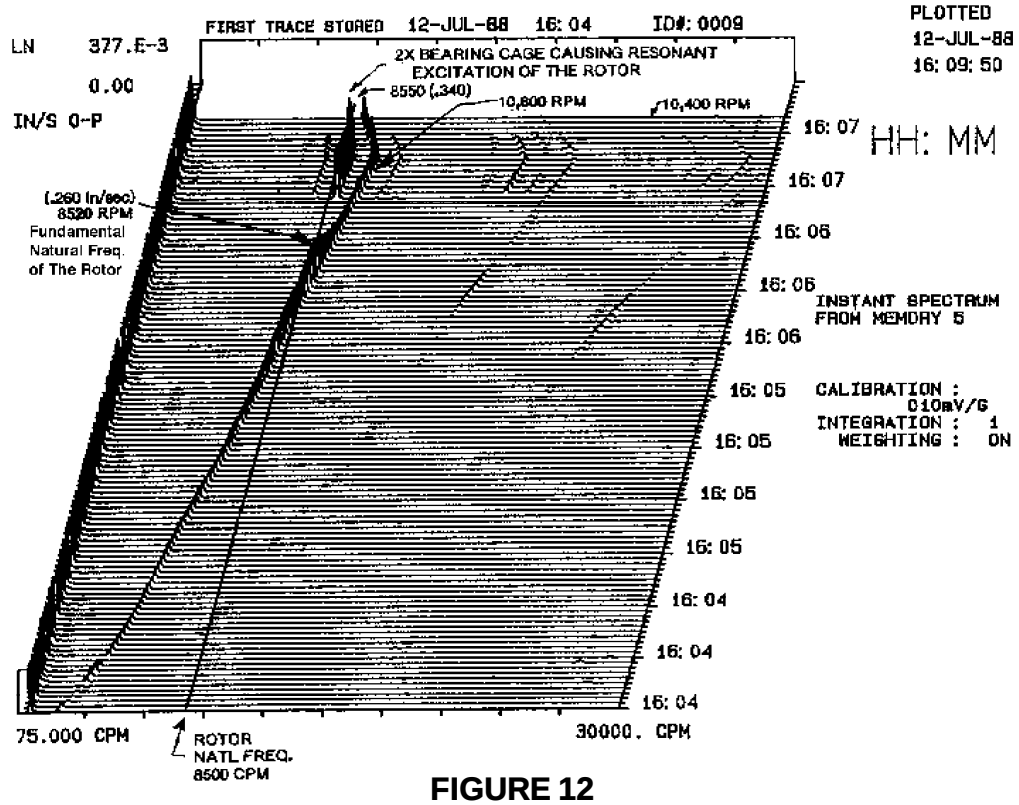

FIGURE 12
CONTROLLED RUNUP TEST AFTER MODIFICATIONS TO GODET ROLL 3A

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REFERENCES

1. *Vibration Technology - 1*, IRD Mechanalysis, Columbus, OH; pages 2-7; 1990.

2. Bently, Donald E.; President and Chairman, Bently Nevada, Inc.; Minden, NV; "Rubs Research - Studies Reveal Physical Phenomena of Rotor Rubs"; *Orbit*, Volume 4, No. 3; pages 3-5; October, 1983.

3. Bently, Donald E.; Zimmer, Siegfried; Palmatier, George E. and Muszynska, Agnes; "Interpreting Vibration Information From Rotating Machinery"; *Sound and Vibration Magazine*; Volume 20, No. 2; pages 14-23; February, 1986.

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CHAPTER 6

USE OF VIBRATION SIGNATURE ANALYSIS TO DIAGNOSE MACHINE PROBLEMS

6.0 USE OF VIBRATION SIGNATURE ANALYSIS

Probably the greatest shortcoming in today's predictive maintenance programs is the ability to diagnose the mechanical and electrical problems within the machine that are evidenced in the vibration signatures if the vibration exceeds preset overall and spectral alarm levels. For example, an ever increasing number of plants have PMP data collectors and software and successfully build large databases and capture vibration measurements at great numbers of points. However, surveys prove that less than 15% of such plants know how to properly set overall and spectral alarm levels. Then, even a smaller minority know how to diagnose the array of potential problems from the vibration spectra and related variables (i.e. spike energy) which are faithfully printed out by the PMP software for those points that exceed alarm. Therefore, the overriding purpose of this section is to begin to acquaint the reader on how to diagnose such problems from this data. Much valuable information is contained within these vibration spectra, but is only of use if the analyst can unlock its "secrets".

Therefore, Table 6.0 has been developed to put many of these "secrets" right at the fingertips of the analyst. Several hundred hours of research have gone into the development of this four page diagnostic chart. Please note that this chart not only provides text elaborating on vibration symptoms for various machine problems, but it also includes illustrations of "typical vibration spectra" for each problem covered. In addition, drawings are included to illustrate how phase reacts when such problems are predominant. Table 6.0 represents the best understanding to date of the author on how these problems are best diagnosed, based on approximately 16 years field experience in vibration signature analysis and research on a wide range of articles which have been written on the subject.

There are several key items included in Table 6.0. First, the plots under "TYPICAL SPECTRUM" column reveal invaluable information as to the source of the problem. When looking at such spectra, the analyst should ask questions similar to the following:

1. Which frequencies are present in spectrum and how do they relate to machine operating speed (that is, are the peaks present equal to 1X, 2X, 3X, 5.78X RPM or what)?

2. What are the amplitudes of each peak?

3. How do the frequency peaks relate to one another? (i.e., "2X RPM is much higher than 1X RPM"; "there is a large peak at 7.43X RPM"; "there are large number of operating speed harmonics present"; "there are high amplitude sidebands around gear mesh frequency"; "there are 7200 CPM sidebands around a large peak at 46X RPM"; etc.).

4. Finally, if there are significant amplitude peaks, what exactly is their source ("is 7.43X RPM a bearing defect frequency"; "is the 46X RPM peak equal to the number of rotor bars RPM"?).

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As its column name implies, “TYPICAL SPECTRUM” is meant to be a representative signature for each problem listed in Table 6.0. These spectra are not intended to be all inclusive. For example, referring to “REMARKS” for the Angular Misalignment problem, note that while the typical spectrum for this problem shows high amplitude 1X RPM and 2X RPM peaks in the axial direction, the discussion shows that it is not unusual for either 1X, 2X or 3X RPM to dominate the whole spectrum when angular misalignment is the problem. In addition, it is not unusual for a machine to have two or more problems present at any one time. For example, if a machine simultaneously had both mechanical looseness and rotor unbalance, they each would contribute frequencies to its spectra which might show high 1X RPM in addition to multiple running speed harmonics.

The next column in Table 6.0 is entitled “PHASE RELATIONSHIP”. Information on phase is provided for several of the problem sources listed. Amplitude reveals how much something is vibrating. Frequency relates how many cycles occur per unit of time. Phase completes the picture by showing just how the machine is vibrating. Of great importance, phase is a powerful tool in helping to differentiate which of several problem sources are dominant. For example, there are a large number of problems that generate vibration at 1X and 2X RPM. Using phase, one learns how the machine is vibrating, and in the process, helps zero in on just which problem is present. For example, Table 6.0 shows how phase reacts during the following scenarios:

1. **Force (or “static”) unbalance** is evidenced by nearly identical phase in the radial direction on each bearing of a machine rotor.

2. **Couple unbalance** shows approximately a 180° out-of-phase relationship when comparing the outboard and inboard horizontal, or the outboard and inboard vertical direction phase on the same machine.

3. **Dynamic unbalance** is indicated when the phase difference is well removed from either 0° or 180°, but importantly is nearly the same in the horizontal and vertical directions. That is, the horizontal phase difference could be almost anything between the outboard and inboard bearings; but, the key point is that the vertical phase difference should then be almost identical to the horizontal phase difference (± 30°). For, example, if the horizontal phase difference between the outboard and inboard bearings is 60°, and the dominant problem is dynamic unbalance, the vertical phase difference between these two bearings should be about 60° (± 30°). If the horizontal phase difference varies greatly from the vertical phase difference, this strongly suggests the dominant problem is not unbalance.

4. **Angular misalignment** is indicated by approximately a 180° phase difference across the coupling, with measurements in the axial direction.

5. **Parallel misalignmen**t causes radial direction phase across the coupling to be approximately 180° out of phase with respect to one another.

6. **Bent shaft** causes axial phase on the same shaft of a machine to approach a 180° difference when comparing measurements on the outboard and inboard bearings of the same machine rotor.

7. **Resonance** is shown by a 90° phase change at the point when the forcing frequency coincides with a natural frequency, and approaches a full 180° phase change when the machine passes through the natural frequency (depending on the amount of damping present).

8. **Rotor rub** causes significant, instantaneous changes in phase.

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9. **Mechanical looseness/weakness due to base/frame problems or loose hold-down bolts** is indicated by nearly a 180° phase change when one moves his transducer from the machine foot down to its baseplate and then down to its support base.

10. **Mechanical looseness due to a cracked frame, loose bearing or loose rotor** causes phase to be unsteady with probable widely differing phase measurements from one measurement to the next. The phase measurement may noticeably differ every time you start up the machine, particularly if the rotor itself is loose and rotates on the shaft a few degrees with each startup.

Often, even though phase measurement capability is now offered by most data collectors, users do not use this powerful tool. If not used, this will severely limit the diagnostic capabilities of any program. However, currently it would be impractical to make phase measurements on all machinery during regular PMP surveys. Its greatest use comes into play when performing diagnostics on machines which have developed high vibration at 1X, 2X or 3X RPM, requiring investigation to detect the predominate cause(s) prior to taking corrective actions.

Note that "PHASE RELATIONSHIP" is illustrated in each of the first 8 problems of Table 6.0 since it is primarily with these problems that phase can be used to differentiate which problem(s) dominate. Phase is then discussed in many of the remarks for the remaining problems in Table 6.0, although it is not illustrated.

Finally, a remarks column is included in Table 6.0 to provide further explanatory information on machine problem symptoms and diagnostics. For example, there is a warning under the remarks column for the "bent shaft" problem source to be sure and account for transducer orientation when taking axial phase measurements.

It is hoped that this illustrated chart will help users in diagnosing a wide variety of machine problems. Further information is now being researched and field tested which may soon be added to the diagnostic chart as we constantly learn more and more about how machines react when subjected to a whole series of problems and how we can "read" these reactive responses via diagnostic techniques.

Following on the next pages will be separate discussions on each of the problems outlined in Table 6.0. Later, real-world case histories will be presented giving real-world examples of each of these problems.

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TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

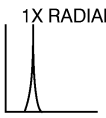
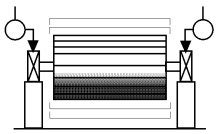
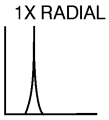
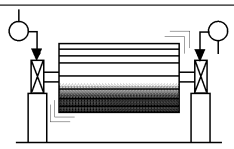
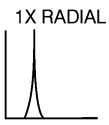
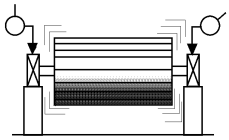
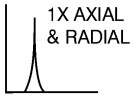
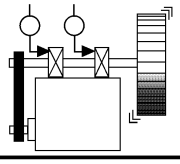
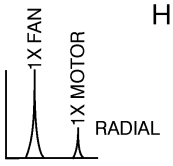
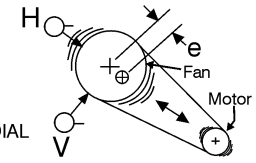
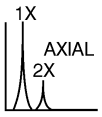
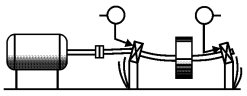
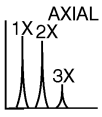
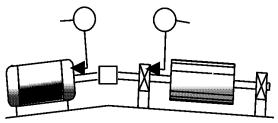
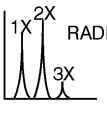
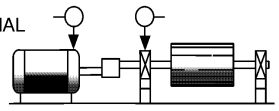
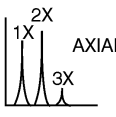
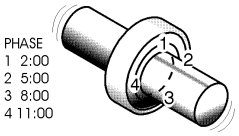
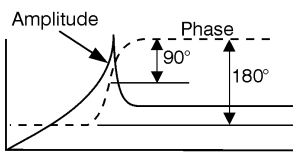
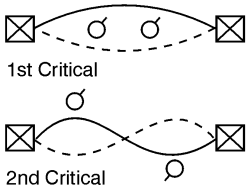
PROBLEM SOURCE	TYPICAL SPECTRUM	PHASE RELATIONSHIP	REMARKS
MASS UNBALANCE **A. FORCE UNBALANCE**			Force Unbalance will be in-phase and steady. Amplitude due to unbalance will increase by the square of speed below first rotor critical (a 3X speed increase = 9X higher vibration). 1X RPM always present and normally dominates spectrum. Can be corrected by placement of only one balance correction weight in one plane at Rotor center of gravity (CG). Approx. 0° phase difference should exist between OB & IB horizontals, as well as between OB & IB verticals. Also, approx. 90° phase difference between horizontal & vertical readings usually occurs on each bearing of unbalanced rotor (±30°).
B. COUPLE UNBALANCE			Couple Unbalance results in 180° out-of-phase motion on same shaft. 1X RPM always present and normally dominates spectrum. Amplitude varies with square of increasing speed below first rotor critical speed. May cause high axial vibration as well as radial. Correction requires placement of balance weights in at least 2 planes. Note that approx. 180° phase difference should exist between OB & IB horizontals, as well as between OB & IB verticals. Also, approx. a 90° difference between the horizontal & vertical phase readings on each bearing usually occurs (±30°)
C. DYNAMIC UNBALANCE			Dynamic Unbalance is the dominant type of unbalance found and is a combination of both force and couple unbalance. 1X RPM dominates the spectrum, and truly requires 2 plane correction. Here, the radial phase difference between outboard and inboard bearings can range anywhere from 0° to 180°. However, the horizontal phase difference should closely match the vertical phase difference, when comparing outboard and inboard bearing measurements (±30°). Secondarily, if unbalance predominates, roughly a 90° phase difference usually results between the horizontal and vertical readings on each bearing (±40°).
D. OVERHUNG ROTOR UNBALANCE			Overhung Rotor Unbalance causes high 1X RPM in both Axial and Radial directions. Axial readings tend to be in-phase whereas radial phase readings might be unsteady. However, the horizontal phase differences will usually match the vertical phase differences on the unbalanced rotor (±30°). Overhung rotors have both force and couple unbalance, each of which will likely require correction. Thus, correction weights will most always have to be placed in 2 planes to counteract both force and couple unbalance.
ECCENTRIC ROTOR			Eccentricity occurs when center of rotation is offset from geometric centerline of a pulley, gear, bearing, motor armature, etc. Largest vibration occurs at 1X RPM of eccentric component in a direction thru centerlines of the two rotors. Comparative horizontal and vertical phase readings usually differ either by 0° or by 180° (each of which indicate straight-line motion). Attempts to balance eccentric rotors often result in reducing vibration in one radial direction, but increasing it in the other radial direction (depending on amount of eccentricity).
BENT SHAFT			Bent shaft problems cause high axial vibration with axial phase differences tending towards 180° on the same machine component. Dominant vibration normally occurs at 1X if bent near shaft center, but at 2X if bent near the coupling. (Be careful to account for transducer orientation for each axial measurement if you reverse probe direction.) Use dial indicators to confirm bent shaft.
MISALIGNMENT **A. ANGULAR MISALIGNMENT**			Angular Misalignment is characterized by high axial vibration, 180° out-of-phase across the coupling. Typically will have high axial vibration with both 1X and 2X RPM. However, not unusual for either 1X, 2X or 3X to dominate. These symptoms may also indicate coupling problems as well. Severe angular misalignment may excite many 1X RPM harmonics. Unlike Mechanical Looseness Type 3, these multiple harmonics do not typically have a raised noise floor on the spectra.
B. PARALLEL MISALIGNMENT			Offset Misalignment has similar vibration symptoms to Angular, but shows high radial vibration which approaches 180° out-of-phase across coupling. 2X often larger than 1X, but its height relative to 1X is often dictated by coupling type and construction. When either Angular or Radial Misalignment becomes severe, they can generate either high amplitude peaks at much higher harmonics (4X-8X), or even a whole series of high frequency harmonics similar in appearance to mechanical looseness. Coupling type and material will often greatly influence the entire spectrum when misalignment is severe. Does not typically have raised noise floor.
C. MISALIGNED BEARING COCKED ON SHAFT			Cocked Bearing will generate considerable axial vibration. Will cause Twisting Motion with approximately 180° phase shift top to bottom and/or side to side as measured in axial direction on same bearing housing. Attempts to align coupling or balance the rotor will not alleviate problem. Bearing usually must be removed and correctly installed.
RESONANCE			Resonance occurs when a Forcing Frequency coincides with a System Natural Frequency, and can cause dramatic amplitude amplification, which might result in premature, or even catastrophic failure. This may be a natural frequency of the rotor, but can often originate from support frame, foundation, gearbox or even drive belts. If a rotor is at or near resonance, it can be almost impossible to balance due to the great phase shift it experiences (90° at resonance; nearly 180° when passes thru). Often requires changing natural frequency to a higher or lower frequency. Natural Frequencies do not generally change with a change in speed which helps facilitate their identification (unless on a large plain bearing machine or on a rotor which has significant overhang).

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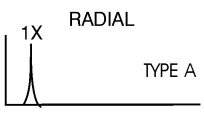
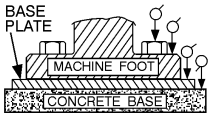
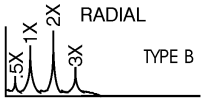
TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

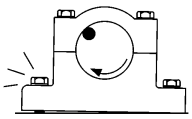
PROBLEM SOURCE	TYPICAL SPECTRUM	PHASE RELATIONSHIP	REMARKS

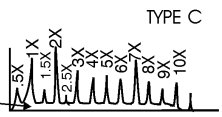
MECHANICAL LOOSENESS

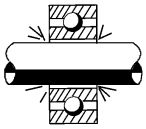






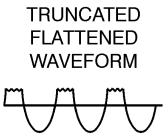


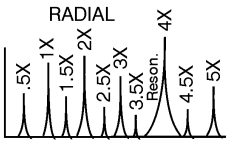




Mechanical Looseness is indicated by either Type A, B or C vibration spectra. ***Type A*** is caused by Structural looseness/weakness of machine feet, baseplate or foundation; also by deteriorated grouting, loose hold-down bolts at the base; and distortion of the frame or base (i.e., soft foot). Phase analysis may reveal approx. 90° to 180° phase difference between vertical measurements on bolt, machine foot, baseplate or base itself.
Type B is generally caused by loose pillowblock bolts, cracks in frame structure or in bearing pedestal.
Type C is normally generated by improper fit between component parts which will cause many harmonics due to nonlinear response of loose parts to dynamic forces from rotor. Causes a truncation of time waveform and a raised noise floor in the spectrum. Type C is often caused by a bearing liner loose in its cap, a bearing loose and turning on its shaft, excessive clearance in either a sleeve or rolling element bearing, a loose impeller on a shaft, etc. Type C Phase is often unstable and may vary widely from one measurement to next, particularly if rotor shifts position on shaft from one startup to next. Mechanical Looseness is often highly directional and may cause very different readings when comparing levels at 30° increments in radial direction all the way around one bearing housing. Also, note that looseness will often cause subharmonic multiples at exactly 1/2 or 1/3X RPM (.5X, 1.5X, 2.5X, etc.).

ROTOR RUB

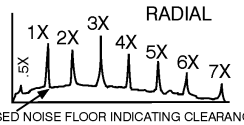




Rotor Rub produces similar spectra to Mechanical Looseness when rotating parts contact stationary components. Rub may be either partial or throughout the entire shaft revolution. Usually generates a series of frequencies, often exciting one or more resonances. Often excites integer fraction subharmonics of running speed (1/2, 1/3, 1/4, 1/5,...1/n), depending on location of rotor natural frequencies. Rotor rub can excite many high frequencies (similar to wide-band noise when chalk is drug along a blackboard). It can be very serious and of short duration if caused by shaft contacting bearing babbitt. A full annular rub throughout an entire shaft revolution can induce "reverse precession" with the rotor whirling at critical speed in a direction opposite shaft rotation (inherently unstable which can lead to catastrophic failure).

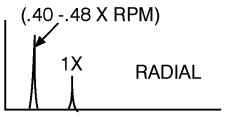
JOURNAL BEARINGS

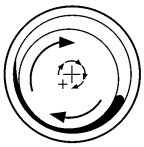
A. WEAR/CLEARANCE PROBLEMS



Latter stages of journal bearing wear are normally evidenced by presence of whole series of running speed harmonics (up to 10 or 20). Wiped journal bearings often will allow high vertical amplitudes compared to horizontal, but, may show only one pronounced peak at 1X RPM. Journal bearings with excessive clearance may allow a minor unbalance and/or misalignment to cause high vibration which would be much lower if bearing clearances were set to spec.

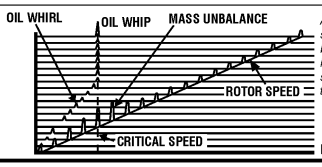
B. OIL WHIRL INSTABILITY





Oil Whirl instability occurs at .40 - .48X RPM and is often quite severe. Considered excessive when amplitude exceeds 40% of bearing clearances. Oil Whirl is an oil film excited vibration where deviations in normal operating conditions (attitude angle and eccentricity ratio) cause oil wedge to "push" shaft around within bearing. Destabilizing force in direction of rotation results in a whirl (or forwards precession). Oil Whirl is unstable since it increases centrifugal forces which increase whirl forces. Can cause oil to no longer support shaft and can become unstable when whirl frequency coincides with a rotor natural frequency. Changes in oil viscosity, lube pressure and external preloads can affect oil whirl.

C. OIL WHIP INSTABILITY

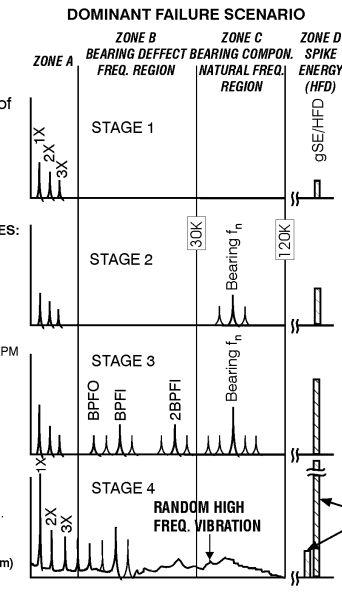


A spectral map showing Oil Whirl becoming Oil Whip instability as shaft speed passes thru 2X critical.

Oil Whip may occur if machine operated at or above 2X rotor critical frequency. When rotor brought up to twice critical speed, whirl will be very close to rotor critical and may cause excessive vibration that oil film may no longer be capable of supporting. Whirl speed will actually "lock onto" rotor critical and this peak will not pass through it even if machine is brought to higher and higher speeds. Produces a lateral forward precessional subharmonic vibration at rotor critical frequency. Inherently unstable which can lead to catastrophic failure.

ROLLING ELEMENT BEARINGS

(4 Failure Stages)



f_n = Natural Frequencies of Installed Bearing Components and Support Structure

BEARING DEFECT FREQUENCIES:

$$BPFI = \frac{N_b}{2}\left(1+\frac{B_d}{P_d}\cos\theta\right) \times RPM$$

$$BPFO = \frac{N_b}{2}\left(1-\frac{B_d}{P_d}\cos\theta\right) \times RPM$$

$$BSF = \frac{P_d}{2B_d}\left[1-\left(\frac{B_d}{P_d}\right)^2(\cos\theta)^2\right] \times RPM$$

$$FTF = \frac{1}{2}\left(1-\frac{B_d}{P_d}\cos\theta\right) \times RPM$$

Where:
BPFI = Inner Race Frequency
BPFO = Outer Race Frequency
BSF = Ball Spin Frequency
FTF = Fund. Train (Cage) Freq.
N_b = Number of Balls or Rollers
B_d = Ball/Roller Diameter (in or mm)
P_d = Bearing Pitch Diameter (in or mm)
θ = Contact Angle (degrees)

4 ROLLING ELEMENT BEARING FAILURE STAGES

STAGE 1: Earliest indications of bearing problems appear in ultrasonic frequencies ranging from about 250,000 - 350,000 Hz; later, as wear increases, usually drops to approximately 20,000 - 60,000 Hz (1,200,000 - 3,600,000 CPM). These are frequencies evaluated by Spike Energy (gSE), HFD(g) and Shock Pulse (dB). For example, spike energy may first appear at about .25 gSE in Stage 1 (actual value depending on measurement location and machine speed). Acquiring high frequency enveloped spectra confirms whether or not bearing is in Failure Stage 1.

STAGE 2: Slight bearing defects begin to "ring" bearing component natural frequencies (f_n) which predominantly occur in 30K - 120K CPM range. Such natural frequencies may also be resonances of bearing support structures. Sideband frequencies appear above and below natural frequency peak at end of Stage 2. Overall spike energy grows (for example, from .25 to .50 gSE).

STAGE 3: Bearing defect frequencies and harmonics appear. When wear progresses, more defect frequency harmonics appear and number of sidebands grow, both around these and bearing component natural frequencies. Overall spike energy continues to increase (for example, from .5 to over 1 gSE). Wear is now usually visible and may extend throughout periphery of bearing, particularly when many well formed sidebands accompany bearing defect frequency harmonics. High frequency demodulated and enveloped spectra help confirm Stage III. ***Replace bearings now! (independent of bearing defect frequency amplitudes in vibration spectra).***

STAGE 4: Towards the end, amplitude of 1X RPM is even effected. It grows, and normally causes growth of many running speed harmonics. Discrete bearing defect and component natural frequencies actually begin to "disappear" and are replaced by random, broadband high frequency "noise floor". In addition, amplitudes of both high frequency noise floor and spike energy may in fact decrease; but just prior to failure, spike energy and HFD will usually grow to excessive amplitudes.

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TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

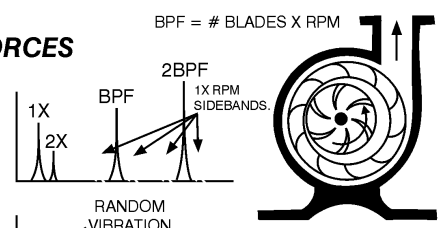
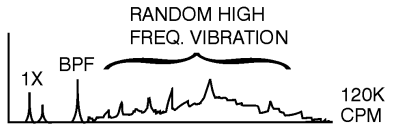
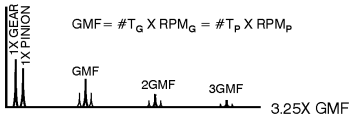
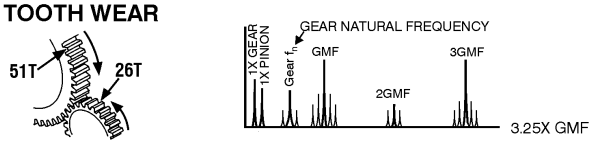
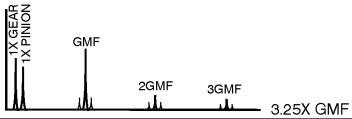
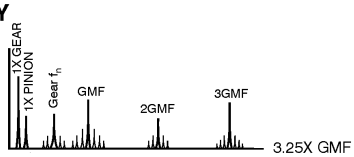
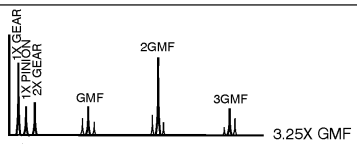
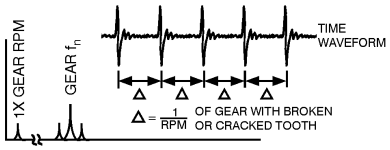
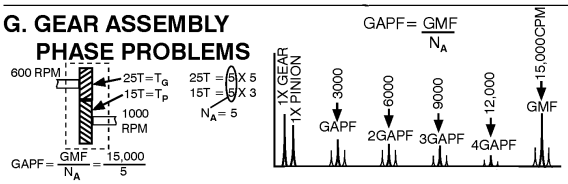
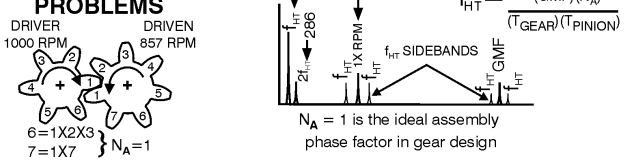
PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS
HYDRAULIC AND AERODYNAMIC FORCES		
A. BLADE PASS & VANE PASS		Blade Pass Frequency (BPF) = No. of Blades (or Vanes) X RPM. This frequency is inherent in pumps, fans and compressors, and normally does not present a problem. However, large amplitude BPF (and harmonics) can be generated in pump if gap between rotating vanes and stationary diffusers is not equal all the way around. Also, BPF (or harmonic) sometimes can coincide with a system natural frequency causing high vibration. High BPF can be generated if impeller wear ring seizes on shaft, or if welds fastening diffuser vanes fail. Also, high BPF can be caused by abrupt bends in pipe (or duct), obstructions which disturb flow, damper settings or if pump or fan rotor is positioned eccentrically within housing.
B. FLOW TURBULENCE		Flow Turbulence often occurs in blowers due to variations in pressure or velocity of the air passing thru the fan or connected ductwork. This flow disruption causes turbulence which will generate random, low frequency vibration, typically in the range of 50 to 2000 CPM. If surging occurs within a compressor, random broadband high frequency vibration can occur. Excessive turbulence can also excite broadband high frequency.
C. CAVITATION		Cavitation normally generates random, higher frequency broadband energy which is sometimes superimposed with blade pass frequency harmonics. Normally indicates insufficient suction pressure (starvation). Cavitation can be quite destructive to pump internals if left uncorrected. It can particularly erode impeller vanes. When present, it often sounds as if "gravel" is passing thru pump. Cavitation is usually caused by insufficient inlet flow. Can occur during one survey, and be absent the next survey (if changes in suction valve settings are made).
GEARS		
A. NORMAL SPECTRUM		Normal Spectrum shows Gear & Pinion Speeds, along with Gear Mesh Frequency (GMF) and very small GMF harmonics. GMF harmonics commonly will have running speed sidebands around them. All peaks are of low amplitude, and no natural frequencies of gears are excited. F_{MAX} recomended at 3.25X GMF (minimum) when # teeth are known. If tooth count is not known, set F_{MAX} at 200X RPM on each shaft.
B. TOOTH WEAR		Key indicator of Tooth Wear is excitation of Gear Natural Frequency (f_n), along with sidebands around it spaced at the running speed of the bad gear. Gear Mesh Frequency (GMF) may or may not change in amplitude, although high amplitude sidebands and number of sidebands surrounding GMF usually occur when wear is noticeable. Sidebands may be better wear indicator than GMF frequencies themselves. Also, high amplitudes commonly occur at either 2XGMF or at 3XGMF (esp. 3XGMF), even when GMF amplitude is acceptable.
C. TOOTH LOAD		Gear Mesh Frequencies are often very sensitive to load. High GMF amplitudes do not necessarily indicate a problem, particularly if sideband frequencies remain low level, and no gear natural frequencies are excited. Each Analysis should be performed with system at maximum operating load for meaningful spectral comparisons.
D. GEAR ECCENTRICITY AND BACKLASH		Fairly high amplitude sidebands around GMF harmonics often suggest gear eccentricity, backlash, or non-parallel shafts which allow the rotation of one gear to "modulate" either the GMF amplitude or the running speed of the other gear. The gear with the problem is indicated by the spacing of the sideband frequencies.Also, 1X RPM level of eccentric gear will normally be high if eccentricity is the dominant problem. Improper backlash normally excites GMF harmonics and Gear Natural Frequency, both of which will be sidebanded at 1X RPM. GMF amplitudes will often decrease with increasing load if backlash is the problem.
E. GEAR MISALIGNMENT		Gear Misalignment almost always excites second order or higher GMF harmonics which are sidebanded at running speed. Often will show only small amplitude 1X GMF, but much higher levels at 2X or 3X GMF. Important to set F_{MAX} high enough to capture at least 3 GMF harmonics. Also, sidebands around 2XGMF will often be spaced at 2X RPM. Note that sideband amplitudes often are not equal on left and right side of GMF and GMF harmonics due to the tooth misalignment. Causes uneven wear pattern.
F. CRACKED/BROKEN TOOTH		A Cracked or Broken Tooth will generate a high amplitude at 1X RPM of this gear only in the time waveform, plus it will excite gear natural frequency (f_n) sidebanded at its running speed. It is best detected in Time Waveform which will show a pronounced spike every time the problem tooth tries to mesh with teeth on the mating gear. Time between impacts (Δ) will correspond to 1/RPM of gear with the problem. Amplitudes of Impact Spikes in Time Waveform often will be 10X to 20X higher than that at 1X RPM in the FFT!
G. GEAR ASSEMBLY PHASE PROBLEMS	 GAPF= 3000 CPM = 0.20X GMF **(FRACTIONAL GMF)**	Gear Assembly Phase Freq. (GAPF) can result in Fractional Gear Mesh Frequencies (if N_A>1). It literally means (T_G/N_A) gear teeth will contact (T_P/N_A) pinion teeth and will generate N_A wear patterns, where N_A in a given tooth combination equals the product of prime factors common to the number of teeth on the gear and pinion (N_A = Assembly Phase Factor). GAPF (or harmonics) can show up right from the beginning if there were manufacturing problems. Also, its sudden appearance in a periodic survey spectrum can indicate damage if contaminate particles pass through the mesh, resulting in damage to the teeth in mesh at the time of ingestion just as they enter and leave meshing or that gears have been reoriented.
H. HUNTING TOOTH PROBLEMS	 f_{HT} = $\frac{(6X\ 1000)(1)}{(6)(7)}$ = $\frac{1000}{7}$ = 143 CPM (One Pulse Per 7 Pinion Revolutions)	Hunting Tooth Frequency (f_{HT}) occurs when faults are present on both the gear and pinion which might have occurred during the manufacturing process, due to mishandling, or in the field. It can cause quite high vibration, but since it occurs at low frequencies predominately less than 600 CPM, it is often missed. A gear set with this tooth repeat problem normally emits a "growling" sound from the drive. The maximum effect occurs when the faulty pinion and gear teeth both enter mesh at the same time (on some drives, this may occur only 1 of every 10 to 20 revolutions, depending on the f_{HT} formula). Note that T_{GEAR} and T_{PINION} refer to number of teeth on the gear and pinion, respectively. N_A is the Assembly Phase Factor defined above. Will often modulate both GMF and Gear RPM peaks.

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TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

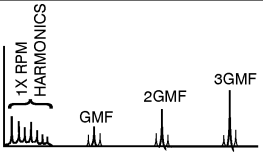
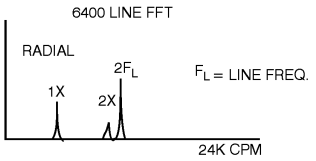
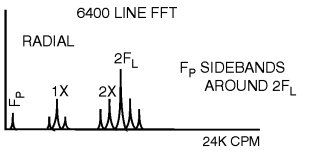
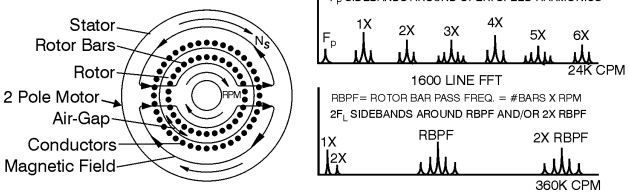
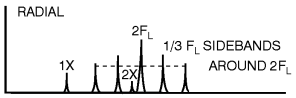
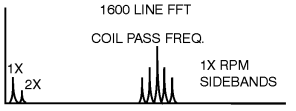
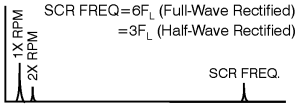
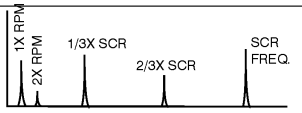
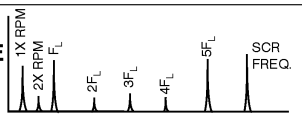
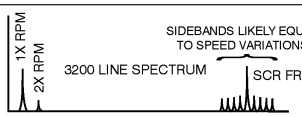
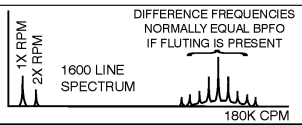
PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS
GEARS (CONTINUED)		
I. LOOSE BEARING FIT		Excessive Clearance of bearings supporting the gears can not only excite many running speed harmonics, but will often cause high amplitude response at GMF, 2GMF and/or 3GMF. These high GMF amplitudes are actually a response to, and not the cause of, looseness within the bearings supporting the gearing. Such excessive clearance can be caused either by extensive bearing wear or by improper bearing fit onto the journal during installation. Left uncorrected, it can cause excessive gear wear and damage to other components.
AC INDUCTION MOTORS		
A. STATOR ECCENTRICITY, SHORTED LAMINATIONS OR LOOSE IRON		Stator problems generate high vibration at 2X line frequency ($2F_L$). Stator eccentricity produces uneven stationary air gap between rotor and stator which produces very directional vibration. Differential Air Gap should not exceed 5% for induction motors and 10% for synchronous motors. Soft foot and warped bases can produce an eccentric stator. Loose iron is due to stator support weakness or looseness. Shorted stator laminations can cause uneven, localized heating which can distort the stator itself. This produces thermally-induced vibration which can significantly grow with operating time causing stator distortion and static air gap problems.
B. ECCENTRIC ROTOR (Variable Air Gap) F_L = Electrical Line Freq. N_S = Synch. Speed = $\frac{120F_L}{P}$ F_S = Slip Freq. = N_S – RPM F_P = Pole Pass Freq. = F_S X P P = #Poles		Eccentric Rotors produce a rotating variable air gap between the rotor and stator which induces pulsating vibration (normally between $2F_L$ and closest running speed harmonic). Often requires "zoom" spectrum to separate $2F_L$ and running speed harmonic. Eccentric rotors generate $2F_L$ surrounded by Pole Pass frequency sidebands (F_P), as well as F_P sidebands around running speed. F_P appears itself at low frequency (Pole Pass Frequency = Slip Frequency X #Poles). Common values of F_P range from about 20 to 120 CPM (0.3 - 2.0 Hz). Soft foot or misalignment often induces a variable air gap due to distortion (actually a mechanical problem; not electrical).
C. ROTOR PROBLEMS		Broken or Cracked rotor bars or shorting rings; bad joints between rotor bars and shorting rings; or shorted rotor laminations will produce high 1X running speed vibration with pole pass frequency sidebands (F_P). In addition, these problems will often generate F_P sidebands around the second, third, fourth and fifth running speed harmonics. Loose or open rotor bars are indicated by 2X line freq. ($2F_L$) sidebands surrounding Rotor Bar Pass Frequency (RBPF) and/or its harmonics (RBPF = Number of Bars X RPM). Often will cause high levels at 2X RBPF, with only a small amplitude at 1X RBPF. Electrically induced arcing between loose rotor bars and end rings will often show high levels at 2X RBPF (with $2F_L$ sidebands); but little or no increase in amplitudes at 1X RBPF.
D. PHASING PROBLEM (Loose Connector)		Phasing problems due to loose or broken connectors can cause excessive vibration at 2X Line Freq. ($2F_L$) which will have sidebands around it spaced at 1/3 Line Freq. (1/3 F_L). Levels at $2F_L$ can exceed 1.0 in/sec if left uncorrected. This is particularly a problem if the defective connector is only sporadically making contact. Loose or broken connectors must be repaired to prevent catastrophic failure.
AC SYNCHRONOUS MOTORS **(Loose Stator Coils)**		Loose stator coils in synchronous motors will generate fairly high vibration at Coil Pass Freq. (CPF) which equals the number of stator coils X RPM (#Stator Coils = #Poles X #Coils/Pole). The Coil Pass Frequency will be surrounded by 1X RPM sidebands. Synchronous motor problems may also be indicated by high amplitude peaks at approx. 60,000 to 90,000 CPM, accompanied by $2F_L$ sidebands. Take at least one spectrum up to 90,000 CPM on each motor bearing housing.
DC MOTORS AND CONTROLS **A. NORMAL SPECTRUM**		Many DC Motor and Control Problems can be detected by vibration analysis. Full-wave rectified, motors (6 SCR's) generate a signal at 6X Line Frequency ($6F_L$=360 Hz=21,600 CPM); while half-wave rectified DC motors (3 SCR's) generate 3X Line Freq. ($3F_L$=180 Hz=10,800 CPM). The SCR firing Frequency is normally present in a DC Motor Spectrum, but at low amplitude. Note the absence of other peaks at multiples of F_L.
B. BROKEN ARMATURE WINDINGS, GROUNDING PROBLEMS OR FAULTY SYSTEM TUNING		When DC Motor spectra are dominated by high levels at SCR or 2X SCR, this normally indicates either Broken Motor Windings or Faulty Tuning of the Electrical Control System. Proper tuning alone can lower vibration at SCR and 2X SCR significantly if control problems predominate. High amplitudes at these frequencies would normally be above approximately .10 in/sec, peak at 1 X SCR and about .04 in/sec at 2 X SCR Firing Freq.
C. FAULTY FIRING CARD OR BLOWN FUSE		When one firing card fails to fire, then 1/3 of power is lost, and can cause repeated momentary speed changes in the motor. This can lead to high amplitudes at 1/3X and 2/3X SCR Frequency (1/3X SCR Freq. = 1X F_L for half-wave rectified, but 2X F_L for a full-wave rectified SCR). Caution: Card/SCR configuration should be known before troubleshooting motor (#SCR's, #Firing Cards, etc.).
D. FAULTY SCR, SHORTED CONTROL CARD, LOOSE CONNECTIONS AND/OR BLOWN FUSE		Faulty SCR's, Shorted Control Cards and/or Loose Connections can generate noticeable amplitude peaks at many combinations of line frequency (F_L) and SCR firing frequency. Normally, 1 bad SCR can cause high levels at F_L and/or $5F_L$ in 6 SCR motors. The point to be made is that neither F_L, $2F_L$, $4F_L$ nor $5F_L$ should be present in DC Motor spectra.
E. FAULTY COMPARITOR CARD		Faulty Comparitor Cards cause problems with RPM fluctuation or "hunting". This causes a constant collapsing and regenerating of the magnetic field. These sidebands often approximate the RPM fluctuation and require a high resolution FFT to even detect them. Such sidebands could also be due to generation and regeneration of the magnetic field.
F. ELECTRICAL CURRENT PASSAGE THRU DC MOTOR BEARINGS		Electrically-induced Fluting is normally detected by a series of difference frequencies with the spacing most often at the outer race defect frequency (BPFO), even if such fluting is present on both the outer and inner races. They most often show up in a range centered at about 100,000 to 150,000 CPM. A 180K CPM spectrum with 1600 lines is recommended for detection with measurements on both the OB and IB DC motor bearings.

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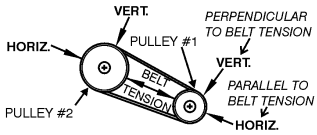
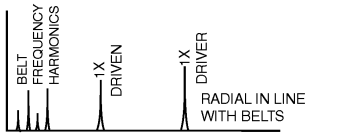
TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS

BELT DRIVE PROBLEMS

A. WORN, LOOSE OR MISMATCHED BELTS





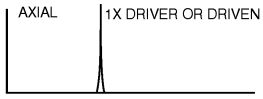
PITCH DIAM$_1$ X RPM$_1$ = PITCH DIAM$_2$ X RPM$_2$

BELT FREQ. = $\frac{3.142 \text{ X PULLEY RPM X PITCH DIAM.}}{\text{BELT LENGTH}}$

TIMING BELT FREQ. = BELT FREQ. X #BELT TEETH
= PULLEY RPM X #PULLEY TEETH

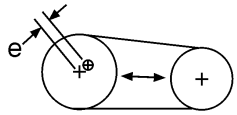
Belt frequencies are below the RPM of either the motor or the driven machine. When they are worn, loose or mismatched, they normally cause 3 to 4 multiples of belt frequency. Often 2X belt freq. is the dominant peak. Amplitudes are normally unsteady, sometimes pulsing with either driver or driven RPM. On timing belt drives, wear or pulley misalignment is indicated by high amplitudes at the Timing Belt Frequency. Chain drives will indicate problems at Chain Pass Frequency which equals #Sprocket Teeth X RPM.

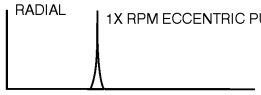
B. BELT/PULLEY MISALIGNMENT



Misalignment of pulley produces high vibration at 1X RPM predominantly in the axial direction. The ratio of amplitudes of driver to driven RPM depends on where the data is taken, as well as on relative mass and frame stiffness. Often with pulley misalignment, the highest axial vibration on the motor will be at fan RPM, or vice versa. Can be confirmed by phase measurements by setting Phase Filter at RPM of pulley with highest axial amplitude; then compare phase at this particular frequency on each rotor in the axial direction.

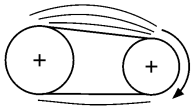
C. ECCENTRIC PULLEYS

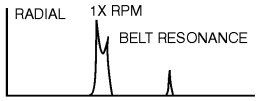




Eccentric pulleys cause high vibration at 1X RPM of the eccentric pulley. The amplitude is normally highest in line with the belts, and should show up on both driver and driven bearings. It is sometimes possible to balance eccentric pulleys by attaching washers to taper-lock bolts. However, even if balanced, the eccentricity will still induce vibration and reversible fatigue stresses in the belt. Pulley eccentricity can be confirmed by phase analysis showing horizontal & vertical phase differences of nearly 0° or 180°.

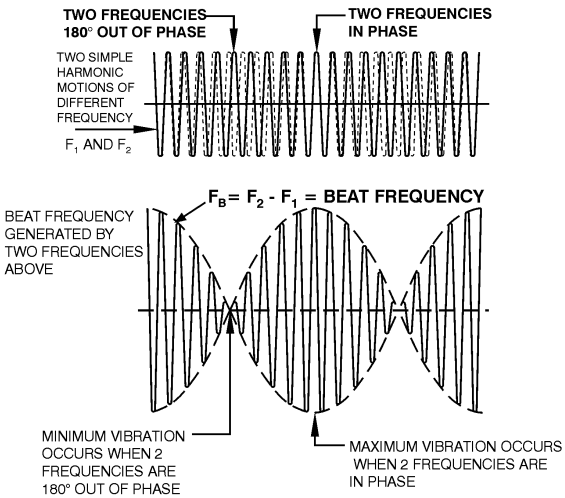
D. BELT RESONANCE





Belt Resonance can cause high amplitudes if the belt natural frequency should happen to approach, or coincide with, either the motor or driven RPM. Belt natural frequency can be altered by changing either the belt tension, belt length or cross section. Can be detected by tensioning and then releasing belt while measuring the response on pulleys or bearings. However, when operating, belt natural frequencies will tend to be slightly higher on the tight side and lower on the slack side.

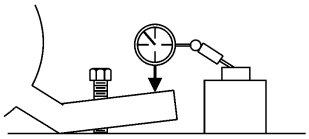
BEAT VIBRATION

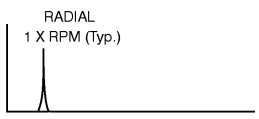

TWO FREQUENCIES 180° OUT OF PHASE
TWO FREQUENCIES IN PHASE
TWO SIMPLE HARMONIC MOTIONS OF DIFFERENT FREQUENCY
F1 AND F2
FB= F2 - F1 = BEAT FREQUENCY
BEAT FREQUENCY GENERATED BY TWO FREQUENCIES ABOVE
MINIMUM VIBRATION OCCURS WHEN 2 FREQUENCIES ARE 180° OUT OF PHASE
MAXIMUM VIBRATION OCCURS WHEN 2 FREQUENCIES ARE IN PHASE

A Beat Frequency is the result of two closely spaced frequencies going into and out of synchronization with one another. The wideband spectrum normally will show one peak pulsating up and down. When you zoom into this peak (lower spectrum below), it actually shows two closely spaced peaks. The difference in these two peaks (F_2-F_1) is the beat frequency which appears itself in the wideband spectrum. The beat frequency is not commonly seen in normal frequency range measurements since it is inherently low frequency, usually ranging from only approximately 5 to 100 CPM.

Maximum vibration will result when the time waveform of one frequency (F_1) comes into phase with the waveform of the other frequency (F_2). Minimum vibration occurs when waveforms of these two frequencies line up 180° out of phase.

SOFT FOOT, SPRUNG FOOT AND FOOT-RELATED RESONANCE





"***Soft Foot***" occurs when a machine's foot or frame deflects greatly when a hold-down bolt is loosened to hand tightness, causing the foot to rise more than approximately .002 - .003 inch. This does not always cause a great vibration increase. However, it can do so if the soft foot affects alignment or motor air gap concentricity.

"***Sprung Foot***" can cause great frame distortion, resulting in increased vibration, force and stress in the frame, bearing housing, etc. This can occur when a hold-down bolt is forceably torqued down on the sprung foot in an attempt to level the foot.

"***Foot-Related Resonance***" can cause dramatic amplitude increases from 5X to 15X or more, as compared with that when the bolt (or combination of bolts) is loosened to hand tightness. When tight, this bolt can notably change the natural frequency of the foot or machine frame itself.

Soft Foot, Sprung Foot or Foot-Related Resonance most often affects vibration at 1X RPM, but can also do so at 2X RPM, 3X RPM, 2X line frequency, blade pass frequency, etc. (particularly Foot-Related Resonance).

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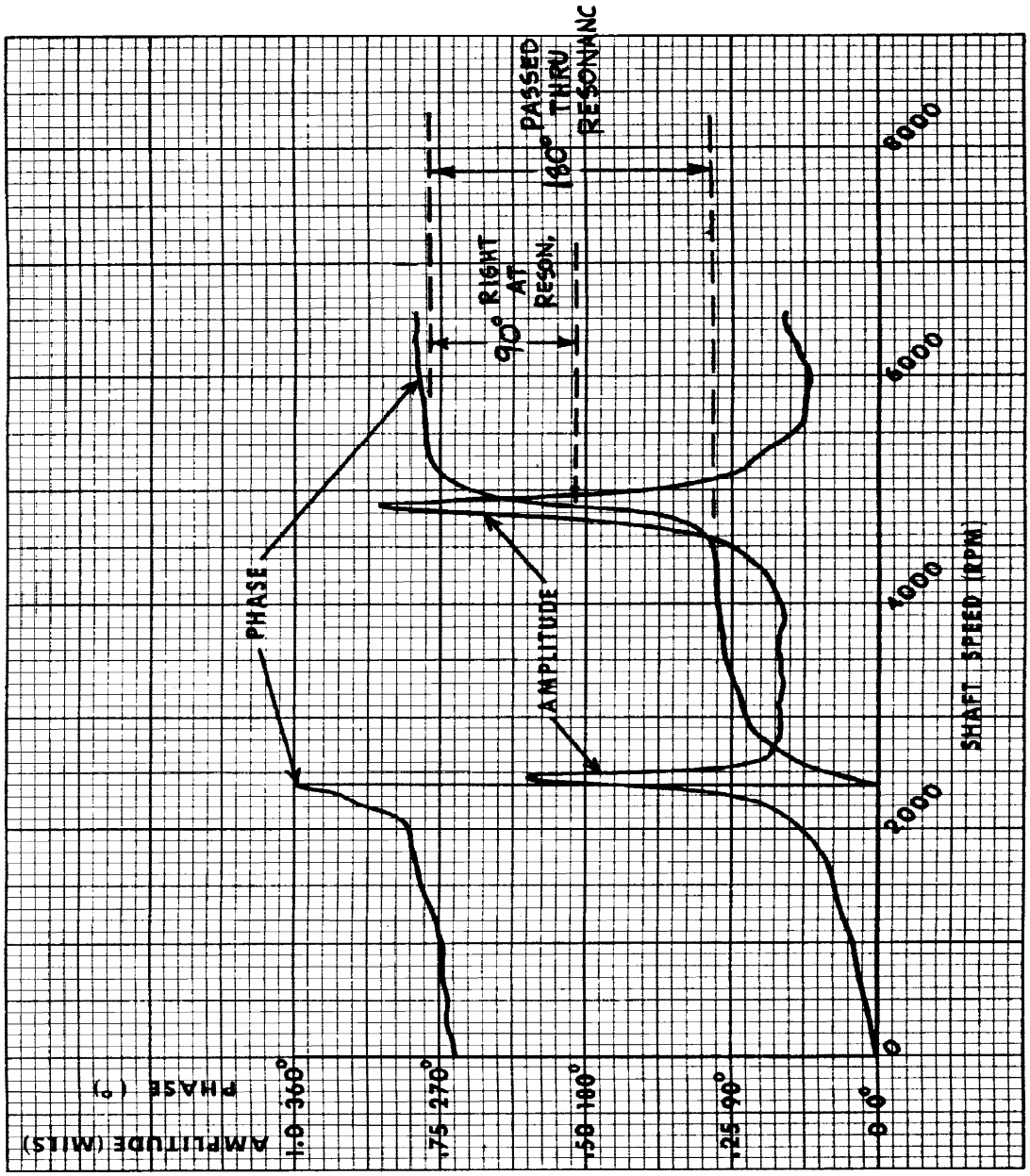

FIGURE 6.0A
PHASE ANALYSIS DIAGRAM BDB-1

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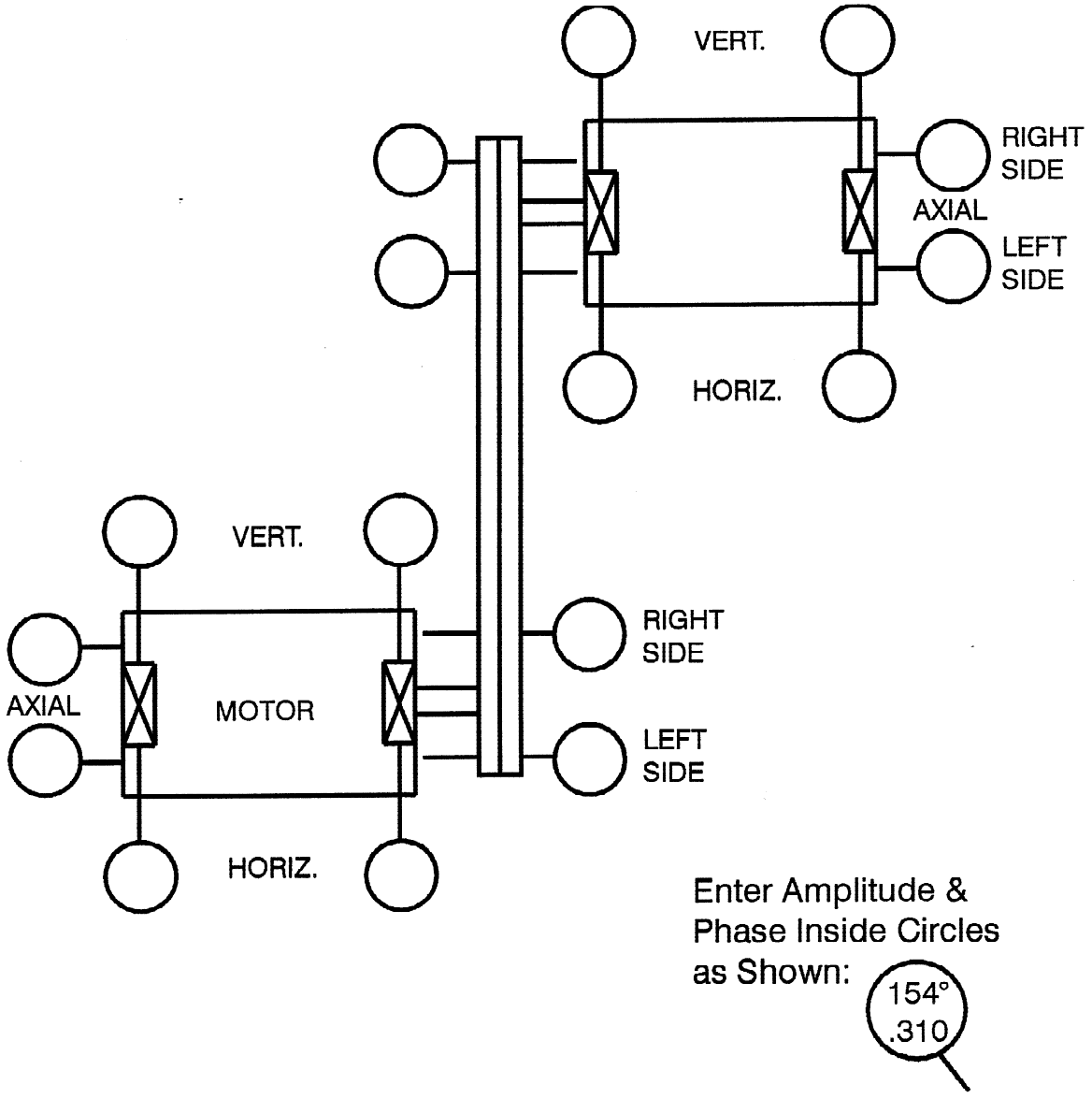

FIGURE 6.0B
PHASE ANALYSIS DIAGRAM BDB-2

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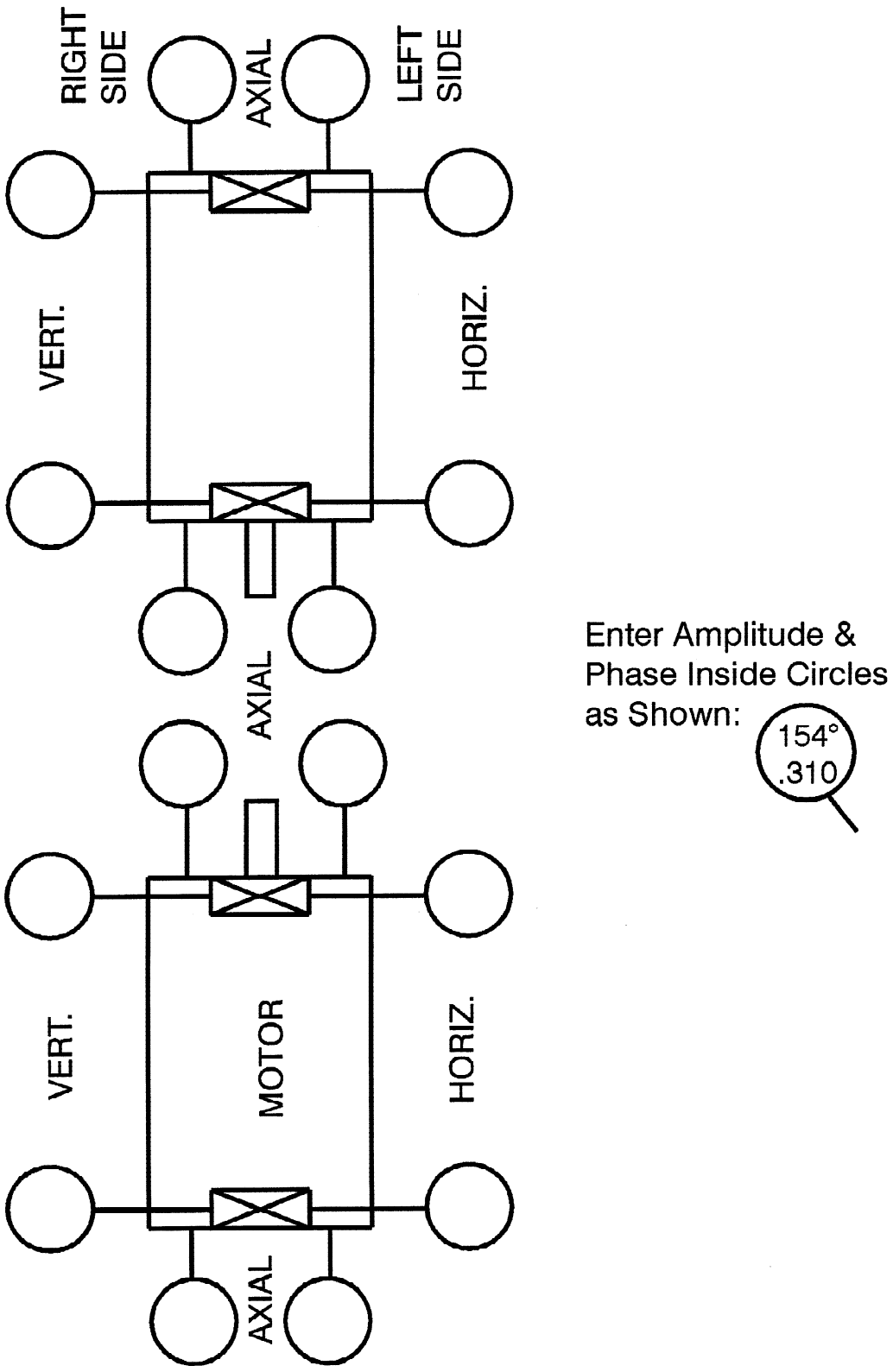

FIGURE 6.0C
PHASE ANALYSIS DIAGRAM DC-1

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6.01 MASS UNBALANCE

Unbalance occurs when the mass centerline does not coincide with the shaft centerline as shown in Figures 6.01A thru 6.01D. Some degree of unbalance exists in all rotors whether they are a cooling tower fan or a precision grinding wheel. The key is to know how much unbalance is acceptable for the particular type of machine at its specific operating speed which will be discussed in Section 6.015 “Allowable Residual Unbalance”.

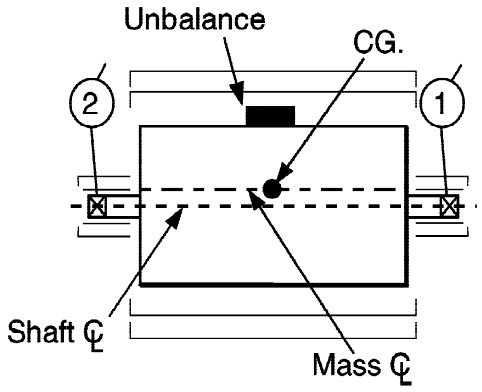

FIGURE 6.01A
FORCE UNBALANCE

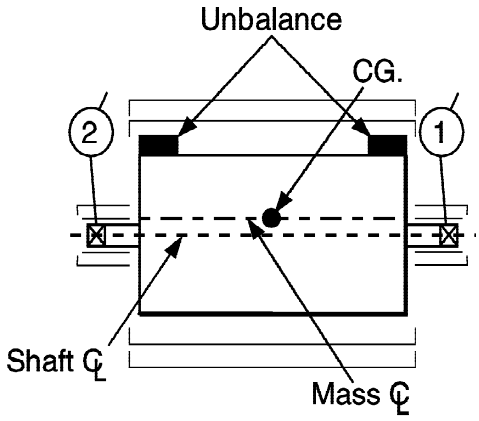

FIGURE 6.01B
FORCE UNBALANCE ALSO

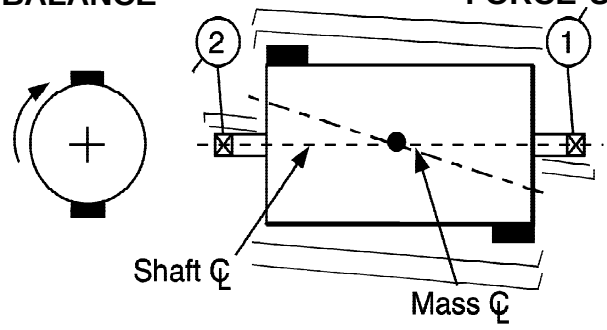

FIGURE 6.01C
COUPLE UNBALANCE

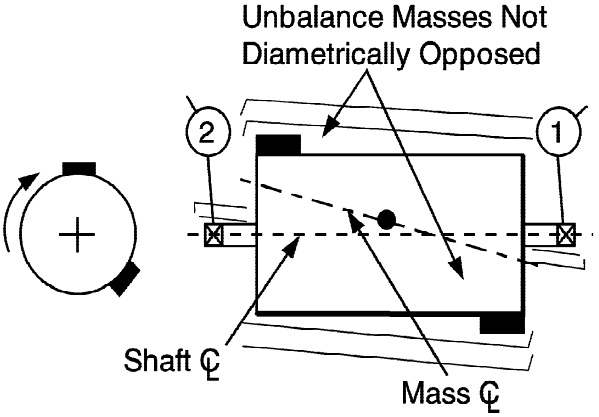

FIGURE 6.01D
DYNAMIC UNBALANCE

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Unbalanced rotors exhibit each of the following characteristics:

1. Unbalance is always indicated by high vibration at 1X RPM of the unbalanced part (but, vibration at 1X RPM is not always unbalance). Normally, this 1X RPM peak will dominate the spectrum.

2. The amplitude at 1X RPM will normally be greater than or equal to 80% of the overall amplitude when the problem is limited to unbalance (may be only 50% to 80% if other problems exist in addition to unbalance).

3. The amplitude of vibration is proportional to how far the mass center is displaced from the shaft center. For example, when operating below the first rotor critical speed, amplitude will vary with the square of RPM (that is, tripling the speed will result in an increase in unbalance vibration by a factor of 9 times).

4. Mass unbalance generates a uniform rotating force which is continually changing direction, but is evenly applied in all radial directions. As a result, the shaft and supporting bearings tend to move in somewhat a circular orbit. However, due to the fact that vertical bearing stiffness is normally higher than that in the horizontal direction, the normal response is a slightly elliptical orbit. Subsequently, horizontal vibration is normally somewhat higher than that in the vertical commonly ranging between 2 and 3 times higher. When the ratio of horizontal to vertical is higher than about 6 to 1, it normally indicates other problems, particularly resonance.

5. When unbalance dominates over other problems, there will normally be about a 90° phase difference between horizontal and vertical directions on a bearing (±30°). Therefore, if there is a high vibration at 1X RPM, but this phase difference is either 0° or close to 180°, it normally points to another problem source such as eccentricity.

6. Probably an even greater indicator of unbalance than the approximately 90° phase shift between horizontal and vertical is the fact that when significant unbalance exists, the horizontal phase difference between outboard and inboard bearings should be close to the difference in phase in the vertical direction. That is, instead of comparing horizontal and vertical phase on the same bearing, compare outboard and inboard horizontal phase difference with outboard and inboard vertical phase difference. For example, please refer to Table A of Figure 6.01E which shows a machine having dominant force unbalance. Note that the horizontal phase difference between the #1 and #2 bearings is about 5° (30° minus 25°) compared to a vertical phase difference of about 10° (120° minus 110°). Similarly, over on the pump, the horizontal phase difference (position 3) is about 10° and the vertical phase difference is about 15°. This is the expected phase response with dominant force unbalance.

7. When unbalance is dominant, radial vibration (horizontal and vertical) will normally be quite much higher than that in the axial direction (except for overhung rotors which will be discussed in Section 6.014).

8. Unbalanced rotors normally exhibit steady and repeatable phase in radial directions. When the rotor is trim balanced, the phase can begin to “dwell” back and forth under a strobe light as you achieve a better and better balance, particularly if other problems are present. However, if there is high unbalance, and other problems are not significant, the phase should be steady and repeatable.

9. The effects of unbalance may sometimes be amplified by resonance. This will be discussed in Section 6.05.

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10. Unbalance can be a great contributor to high looseness vibration. In fact, on a rotor with unbalance and looseness, if it is possible to balance the rotor, this may substantially reduce the looseness vibration although it will often return when even the least little unbalance component returns. Often, it is not even possible to balance rotors having noticeable looseness.

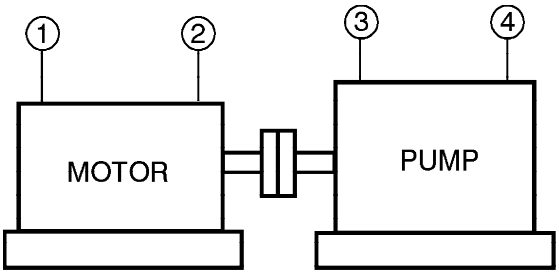

TABLE A.
PHASE AT 1X RPM FOR A MACHINE
WITH DOMINANT FORCE UNBALANCE

DIR.	1	2	3	4
A	60°	*70°	60°	*80°
H	30°	25°	30°	40°
V	120°	110°	120°	135°

* Corrected Phase Values Accounting For 180° Orientation of Accelerometer.

TABLE B.
PHASE AT 1X RPM FOR A MACHINE
WITH DOMINANT COUPLE UNBALANCE

DIR.	1	2	3	4
A	60°	*70°	60°	*80°
H	30°	210°	200°	180°
V	120°	295°	280°	300°

* Corrected Phase Values Accounting For 180° Orientation of Accelerometer.

TABLE C.
PHASE AT 1X RPM FOR A MACHINE
WITH SIGNIFICANT DYNAMIC UNBALANCE

DIR.	1	2	3	4
A	60°	*70°	60°	*80°
H	30°	90°	80°	70°
V	120°	180°	170°	165°

* Corrected Phase Values Accounting For 180° Orientation of Accelerometer.

FIGURE 6.01E
TYPICAL PHASE MEASUREMENTS WHICH WOULD INDICATE EITHER FORCE, COUPLE OR DYNAMIC UNBALANCE

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There are 3 major types of unbalance including force, couple and dynamic unbalance which will be discussed in Sections 6.011, 6.012 and 6.013, respectively, which follow:

6.011 Force Unbalance:

Force unbalance is sometimes known as “static unbalance”. Force unbalance is a condition where the mass centerline is displaced from and parallel to the shaft centerline as shown in Figure 6.01A. This is the type of unbalance that has been classically corrected for many years by placing a fan rotor on knife edges or within its bearings and allowing it to “roll to the bottom”. That is, when the fan wheel is released, if the heavy spot is angularly displaced from the bottom (6:00 position), it would tend to roll to the bottom hopefully ending up in the 6:00 position, assuming the rotor was sufficiently free within its bearings to rotate. So-called correction of this force unbalance was then accomplished by placing a weight opposite this location (or at about 12:00).

Actually, there are two types of force unbalance as shown in Figure 6.01A and Figure 6.01B. In the case of Figure 6.01A, only one heavy spot exists and is located close to the rotor center of gravity (CG). This is corrected by simply placing an equal weight 180° opposite the angular position of the heavy spot. Figure 6.01B likewise illustrates force unbalance even though it shows heavy spots acting on both the outboard and inboard planes (angularly parallel to one another). In this case, it can either be corrected by placing correction weights either at the CG, or by placing equal and opposite weights at each of the two planes (if corrected at the CG, it would of course require double the correction weight in this case).

Characteristics common to force unbalance can be summarized as follows:

1. Approximately the same unbalance forces at 1X RPM are normally present both on the outboard and inboard rotor bearing housings (however, horizontal and vertical responses may differ somewhat depending on the support stiffness in each direction).

2. With pure force unbalance, the outboard horizontal phase will equal the inboard horizontal phase on the same shaft (that is, if the horizontal phase on the outboard bearing were at 6:00, the inboard reading should likewise be about 6:00 since the two shaft ends are moving together).

3. Likewise, the outboard vertical phase should approximately equal the inboard vertical phase on the same shaft.

4. Force unbalance only requires a single plane correction with the counterweight acting through the rotor CG.

5. The difference in horizontal outboard and inboard phase should approximately equal the phase difference in outboard and inboard vertical phase and the phase change across the coupling should be small (less than 60° to 90°) if force unbalance were dominant.

6.012 Couple Unbalance:

Couple unbalance is a condition where the mass centerline axis intersects the shaft centerline axis at the rotor’s center of gravity as shown in Figure 6.01F. Here, a couple is created by equal heavy spots at each end of the rotor, but 180° opposite each other. Significant couple unbalance can introduce severe instability to the rotor causing it to wobble back and forth (like a “seesaw” with the fulcrum at the rotor CG).

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Couple unbalance exhibits each of the following characteristics:

1. In pure couple unbalance, the rotor is statically balanced and will not roll to the bottom when the rotor is placed on knife edges. That is, referring to Figure 6.01C, since the heavy spot at position 1 is equal to that at position 2, this meets the requirement for force or static balance. Still, this rotor will also generate considerable vibration at 1X RPM.

2. Couple unbalance generates high amplitude vibration at 1X RPM on both the outboard and inboard bearing housings, but it may be somewhat higher on one bearing than on the other.

3. Substantial couple unbalance can sometimes generate high axial vibration.

4. The horizontal phase difference between the outboard and inboard bearings will approximate 180° (that is, if the outboard horizontal phase were at 6:00, then the inboard horizontal phase will probably be about 12:00 since the two ends are moving opposite each other in a rocking motion).

5. Similarly, the vertical phase difference between outboard and inboard bearings will approximate 180°.

6. Refer to Table B of Figure 6.01E illustrating how phase should react to couple unbalance. Note the 180° phase difference between position 1 and 2 horizontal (210° - 30°), and the 175° phase difference between position 1 and 2 vertical (295° - 120°). This shows that if the problem is couple unbalance (and not misalignment), both the horizontal and vertical phase differences should roughly be equal to one another - both approximately 180° difference between the outboard and inboard bearings.

6.013 Dynamic Unbalance:

Dynamic unbalance is by far the most common type of unbalance as compared to either purely force or couple unbalance and is defined as “that condition in which the mass centerline is neither parallel to nor intersects the shaft centerline axis”. In essence, dynamic unbalance is a combination of both force and couple unbalance. It requires correction in at least 2 planes perpendicular to the shaft centerline axis.

Dynamic unbalance exhibits each of the following characteristics:

1. Dynamic unbalance generates high vibration at 1X RPM, but the amplitude on the outboard bearing may be somewhat different than that on the inboard bearing housing. Still, they should be within the same order of magnitude, or below about 3 to 1 assuming there are no other significant problems present.

2. Like force and couple unbalance, phase is still steady and repeatable when dynamic unbalance dominates.

3. Although the horizontal phase difference between outboard and inboard bearings could be anything from 0° to 180°, this difference should still approximately equal the vertical phase difference. For example, if the horizontal phase difference was about 60°, the vertical phase difference should likewise be about 60° (±30°) as illustrated in Table C of Figure 6.01E. Here, in this example, notice that the phase difference in both horizontal and vertical directions at positions 1 and 2 is about 60° and that the phase difference across the coupling does not approach 180°. Dynamic unbalance requires correction in at least 2 planes.

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4. Whether or not force or couple balance dominates, the horizontal phase difference at bearings 1 and 2 should approximately equal the vertical phase difference at these two bearings (if the horizontal phase difference is about 150° showing high couple unbalance, the vertical phase difference will approximate 150° as well) **.**

6.014 Overhung Rotor Unbalance:

Figure 6.01F shows an overhung rotor. In this case, the driven rotor is placed outboard of bearings 1 & 2 (rotors which are placed between bearings are known as simply supported rotors). Overhung rotors can cause some interesting vibration symptoms and often can present real problems to the analyst when he attempts to balance one. Overhung rotors display the following characteristics:

1. Overhung rotors can generate large axial forces at 1X RPM which can cause axial vibration to be equal to or greater than radial vibration amplitudes.

2. Overhung rotors often generate a high degree of couple unbalance in addition to force unbalance, both of which must be corrected.

3. Referring to Figure 6.01F, for pure unbalance of an overhung rotor, the axial phase at bearing 1 will approximately equal that at bearing 2 (±30°). Here again, this phase difference depends on how dominant the unbalance problem is as compared to others such as misalignment, resonance, etc.

4. Normally, overhung rotor unbalance can be corrected by first taking care of the force unbalance component which would leave the remainder as couple unbalance with phase differences approaching 180°. The couple component would then require placement of correction weights in 2 planes 180° opposite one another.

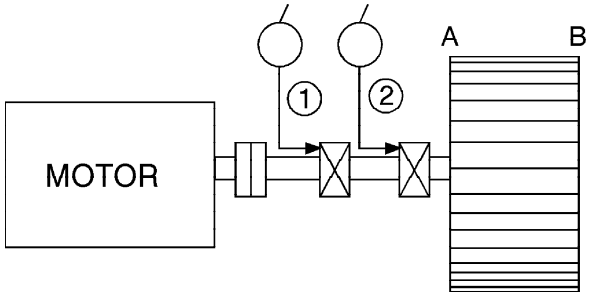

FIGURE 6.01F
BALANCING OF AN OVERHUNG ROTOR

6.0141 Summary of Procedures for Balancing Overhung Rotors

Overhung rotors are machine configurations like that shown in Figure 6.01G where the fan wheel to be balanced is outboard of its two supporting bearings. This configuration is very often found with machines such as blowers, pumps, etc. Because the planes where balance correction weights must be attached are outside the supporting bearings, these rotors will often not respond to standard single and two-plane balancing techniques. In addition, because the unbalance planes are outside the support bearings, even a static unbalance alone will create a couple unbalance proportional to the distance of the unbalance plane from the rotor CG. Therefore, when attempting to balance overhung rotors, the analyst needs to take into account both static and couple unbalance forces, and treat them accordingly.

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When balancing an overhung rotor, one of the two following procedures should be taken:

1. Balancing Overhung Rotors by Classic Single-Plane Static-Couple Method:

Figure 6.01G helps explain methods of balancing overhung rotors. Classically, Bearing A is most sensitive to static unbalance whereas the bearing farthest from the fan wheel to be balanced (Bearing B) is most sensitive to couple unbalance. Since Plane 1 is closest to the rotor center of gravity (CG), static corrections should be made in this plane while measuring the response on Bearing A. On the other hand, measurements should be made on Bearing B when making couple corrections in Plane 2. However, placing a trial weight in Plane 2 will destroy the static balance achieved at Bearing A. Therefore, in order to maintain the static balance at Bearing A, a trial weight placement which will generate a couple must be used. Thus, a trial weight of identical size should be placed in Plane 1 at an angle 180° opposite the trial weight location in Plane 2.

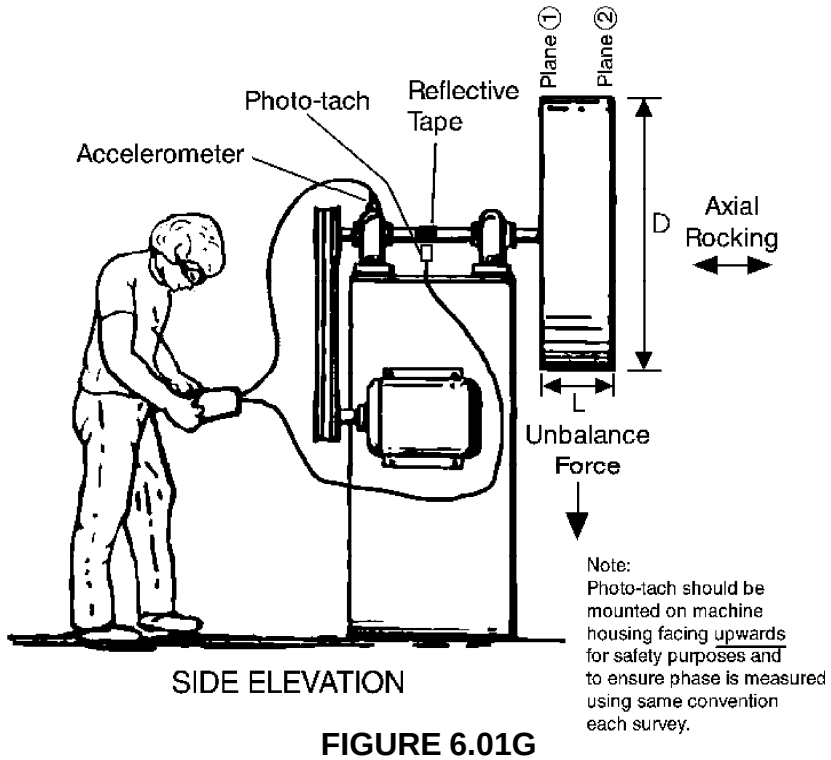

FIGURE 6.01G
FIELD INSTRUMENT SETUP FOR BALANCING OVERHUNG ROTORS

Therefore, either the data collector can be used using single-plane balance software or the single-plane graphic technique previously explained can be successfully employed on many overhung rotors, particularly if the ratio of the rotor length-to-diameter (L/D) is less than approximately .50 (where L is length of the rotating component on which correction weights will be placed and D is the diameter of this component - see Figure 6.01G).

Following below will be a description of this classic single-plane balancing technique for overhung rotors:

a. Set Up Data Collector and/or Spectrum Analyzer Instruments - The data collector, phototach, accelerometer and so forth should be set up as previously described under Section D and Figure 6.01G showing the two-plane balancing procedure. Alternatively, the analyst may wish to employ either a swept-filter analyzer which drives a strobe light (like an IRD 350 or IRD 880), or a spectrum analyzer which will fire a phototach for phase measurement.

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b. Take Initial Measurements - Take initial measurements of 1X RPM amplitude, frequency and phase before adding any trial weights. Measurements should be taken on both the outboard and inboard bearings in both vertical and horizontal directions. The radial direction measurement having the highest amplitude will normally be employed for initial balancing (however, after correcting unbalance in the radial direction, measurements will have to be taken in the other radial direction to ensure amplitudes in it are likewise acceptable).

c. Determine if the Dominant Problem is Either Static or Couple Unbalance - Looking at the amplitude and phase measurements taken on both bearings in the radial and horizontal directions, determine if the problem is dominated by either static or couple unbalance. If phase differences between the outboard and inboard bearing are approximately 140° or more in both the vertical and horizontal directions, the dominant problem will be couple unbalance. On the other hand, if these differences are both anywhere from 0° to approximately 40°, static unbalance is dominant. Of course, phase differences ranging from approximately 40° to 140° are truly dynamic balance once again with a combination of static and couple. If the problem appears to be mostly couple unbalance, use couple unbalance procedures outlined below. However, if the problem appears to be predominantly static or dynamic unbalance, employ static balance procedures. For now, we will assume that the problem is mostly static.

d. Make a Single-Plane Static Balance - Referring to Figure 6.01G, use single-plane techniques taking measurements on Bearing A and placing trial and correction weights in Plane 1.

e. Determine if Resultant Vibration Amplitudes Meet Required Criteria - After completing the single-plane static balance using Plane 1, repeat vibration measurements on both the outboard and inboard bearings in each direction (including axial) and ensure that amplitudes now meet allowable criteria.

f. If Considerable Couple Unbalance Now Remains, Continue With Single-Plane Balance From Bearing B - Overhung rotors often have large cross-effects which means that single-plane balancing from Plane 1 will often cause high vibration over at Bearing B. Therefore, the analyst will perform another single-plane balance, this time making his measurements from Bearing B farthest from the component to be balanced. When he arrives at the single-plane correction weight solution, he should place this weight in Plane 2; and then place an identical size correction weight over in Plane 1 some 180° away from the weight location in Plane 2.

g. Determine if Amplitudes Now Meet all Criteria - After completing the single-plane couple correction, the analyst must again make measurements in horizontal, vertical and axial directions on each bearing and determine that all amplitudes now meet allowable criteria. Often, further balancing must be done at this point beginning with another single-plane balance using Bearing A and Plane 1 which might possibly be followed by another couple balance correction.

h. If Allowable Criteria Cannot be Met in all Three Directions of Each Bearing, Proceed to Two-Plane Balance Procedure Outlined Below - Sometimes, this single plane approach will not successfully reduce amplitudes below allowable criteria in all three directions on each bearing, particularly if the L/D ratio is greater than .50 or if the component to be balanced is located far away from the closest bearing. If this happens, two-plane techniques outlined below will have to be taken.

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2. Balancing Overhung Rotors by Classic Two-Plane Static-Couple Method:

Due to the significant cross-effects which are often present in overhung rotors, two-plane balance correction techniques often are more successful than those employing single-plane methods. However, one of the problems with two-plane methods is that it can sometimes be a little confusing on deciding which bearing is the left and which is the right bearing; similarly, which plane is the left and which is the right plane? (Some data collectors refer to these as the near and far planes as opposed to left and right; terminology does not matter - only that the analyst remain consistent in his convention.) Referring to Figure 6.01G, when using two-plane techniques, Bearing A will be considered the bearing closest to the overhung rotor while Bearing B will be closest to the pulley. Similarly, Plane 1 will be on the inboard side of the wheel closest to the bearings whereas Plane 2 will be outboard.

Here again, a static/couple solution will be employed when the two-plane correction weight calculations are completed. Since most overhung rotors are so sensitive to static unbalance, only the static correction weight will be placed when this static/couple solution is obtained. Then, after trim balancing, if considerable couple unbalance remains, the analyst will proceed to correct this as well. He should follow the procedure outlined below:

a. Set Up Instruments as Outlined in Two-Plane Balance Method in Figure 6.01G - Here again, this same procedure can be used with either data collectors, swept-filter analyzers or real-time analyzers. However, if using either a swept-filter or real-time analyzer, the analyst should have a two-plane calculator program that is capable of providing static/couple solutions.

b. Take Initial Measurements on Both Bearings - Here again, 1X RPM amplitude, frequency and phase should be measured in horizontal, vertical and axial directions on both the outboard and inboard bearings.

c. Complete a Two-Plane Balancing Procedure, But Do Not Yet Place Balance Correction Weights - A two-plane balance procedure like that outlined in Section D should be employed, but final correction weights not put in place. Instead, when the trial weights sizes and locations are calculated for each plane, the analyst should ask for a static/ couple solution and should initially only make the static correction. For example, if the static solution called for 1 oz. in Plane 1 whereas the couple solution called for a 2 oz. correction in Planes 1 & 2 180° opposite one another, make only the static correction at this point.

d. Determine if Amplitudes Now Meet Allowable Criteria - After making the static correction in Plane 1, see if amplitudes in all three directions on each bearing are now within compliance with allowable criteria. If not, trim as required. Again, when the two-plane corrections are determined, ask for the static/couple solution and once again, make only the static correction. Most of the time, the problems are resolved at this point. However, if considerable couple unbalance still remains, complete another two-plane procedure again asking for the static/couple solution - this time making the couple correction called for, and not the static correction.

e. Determine if Amplitudes Now Meet Allowable Criteria - After each of the two trials making these static corrections and the single trial making the couple correction, compare amplitudes in horizontal, vertical and axial directions on both the outboard and inboard bearings with allowable criteria. A small percentage of the time, the couple correction will throw the static balance back off. If this is the case, it may require one more static correction before the rotor is successfully balanced.

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6.015 Allowable Residual Unbalance and ISO Balance Quality Grade:

When balancing a rotor, one needs to know to what precision he is required to balance the unit. In reality, it is not enough to simply say that it will be acceptable to balance the machine to a level of .10 in/sec or 1.0 mil at a certain speed. While one rotor may satisfactorily be balanced at such levels, another will not. This was recognized back in the 1950's by experts involved in the balance field. They recognized that the residual unbalance is truly proportional to the amount and radius of the remaining rotor eccentricity as well as the weight of the rotor itself and its operating speed. Therefore, they developed a series of balance tolerances known as ISO Standard No. 1940 on "Balance Quality of Rotating Rigid Bodies". Table 6.01A provides the balance quality grades as per these standards for a whole group of rotor types. Then, Table 6.01B provides the numerical standards for each of the ISO balance quality grades (ISO G-1, ISO G-2.5, ISO G-6.3, etc.). Note that the lower the G tolerance, the more precision the balance quality grade. Also note that it is based on the rotor RPM (horizontal axis) as well as the residual unbalance per pound of rotor weight (vertical axis). Table 6.01C is provided showing these same balance quality grades delineated by bands separating one balance quality grade from another. Table 6.01C also shows common nominal RPM's in the United States (1200, 1800 and 3600 RPM).

When balancing a machine, you first refer to the tabulated information in Table 6.01A to determine to what tolerance you should balance it. For example, if balancing an automobile crankshaft, this falls under ISO G-16 quality grade. On the other hand, fans fall under G-6.3, and grinding-machine drives fall under G-1. Note that the quality grade number itself represents the maximum permissible circular velocity of the rotor center of gravity expressed in millimeters per second (mm/sec). For example, a quality grade G-6.3 corresponds to rotor velocity of 6.3 mm/sec RMS which corresponds to an equivalent .248 in/sec RMS (.351 in/sec peak).

It is the experience of the author that this ISO Standard, which was established in 1966, is a little too conservative, possibly because it might be based on the technology available during that day and time. It is recommended that when using the ISO tolerance, that you should use one quality grade better than that specified for the specific machine you wish to balance (that is, if the standard calls for G-6.3, we would recommend using ISO G-2.5). For example, if balancing a fan wheel, note that Table 6.01A calls for an ISO quality grade G-6.3 for fans. In this case, we would recommend ISO grade G-2.5. Following below will be a procedure on how to determine the allowable residual unbalance, the ISO balance quality grade that you have achieved, and the rotor balance sensitivity:

How to Determine Residual Unbalance Remaining in a Rotor After Balancing

When field balancing, one must know when to determine that the job is complete. He will know this not only when he has achieved low vibration levels, but also when he knows he has balanced the rotor within allowable specifications. To know this, he must determine the residual unbalance remaining in the rotor. This can be accomplished by following the procedure below (refer to Figure 6.01H):

a. Make original measurements of amplitude and phase and graph this to scale on polar coordinate paper. Call this vector the "O" vector.

b. Attach a trial weight and document the trial weight size (oz) and radius (in) to which it is attached. (mr = trial weight size X trial weight radius)

c. After attaching the trial weight, spin the rotor and measure amplitude and phase. Graph this on the polar coordinate paper as the "O + T" vector.

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d. Draw a vector called “T” from the end of vector “O” to the end of vector “O + T”. Vector "T" represents the effect of the trial weight alone. Measure the length of vector "T" to the same scale as that used for vectors "O" and "O + T". Using this scale, determine the equivalent vibration level (mils).

e. Calculate Rotor Sensitivity as per the following equation:

$$\text{Rotor Sensitivity (oz-in/mil)} = \frac{\text{(Trial Wt. Size)(Trial Wt. Radius)}}{\text{Trial Weight Effect}}$$

(Eqn. 6.01A)

f. Calculate Residual Unbalance using Equation 6.01B. If Residual Unbalance not brought within tolerances, trim balance using current correction weight as the trial weight for the trim run. Continue trim balancing until Residual Unbalance is reduced within required balance tolerances:

$$\underset{\text{(oz-in)}}{\text{Residual Unbalance}} = \underset{\text{(oz-in/mil)}}{\text{Rotor Sens.}} \times \underset{\text{(mils)}}{\text{Vib.Ampl. After Bal.}}$$

(Eqn. 6.01B)

Example (see Figure 6.01H):

Given:

Required ISO Balance Quality = G 2.5
Rotor Weight = 100 lb
Rotor Speed = 800 RPM
Amplitude After Balancing = 2.0 mils

Therefore, $\underline{\text{Required } U_{per}} = \underline{\text{1.76 oz-in total}}$ (single-plane balance)

a. Original reading = 10 mils @ 240° = “O” vector.

b. Trial weight of 3 oz is attached in the balance plane at a 6 inch radius (mr = 3 oz X 6 in = 18 oz-in)

c. Trial run reading = 8 mils @ 120° = “O + T” vector

d. Effect of trial weight alone = T = 15.5 mils (from Figure 6.01H)

e. Rotor Sensitivity = $\frac{\text{18 oz-in}}{\text{15.5 mils}} = 1.16 \frac{\text{oz-in}}{\text{mil}}$

f. Residual Unbalance = $(1.16 \frac{\text{oz-in}}{\text{mil}})(2.0 \text{ mils}) = \underline{\text{2.32 oz-in}}$ (not within specs)

Continued balancing and reduced vibration to 1.0 mil

$$\text{Residual Unbalance} = (1.16 \tfrac{\text{oz-in}}{\text{mil}})(1.0 \text{ mil}) = \underline{\text{1.16 oz-in}} \text{ (in compliance)}$$

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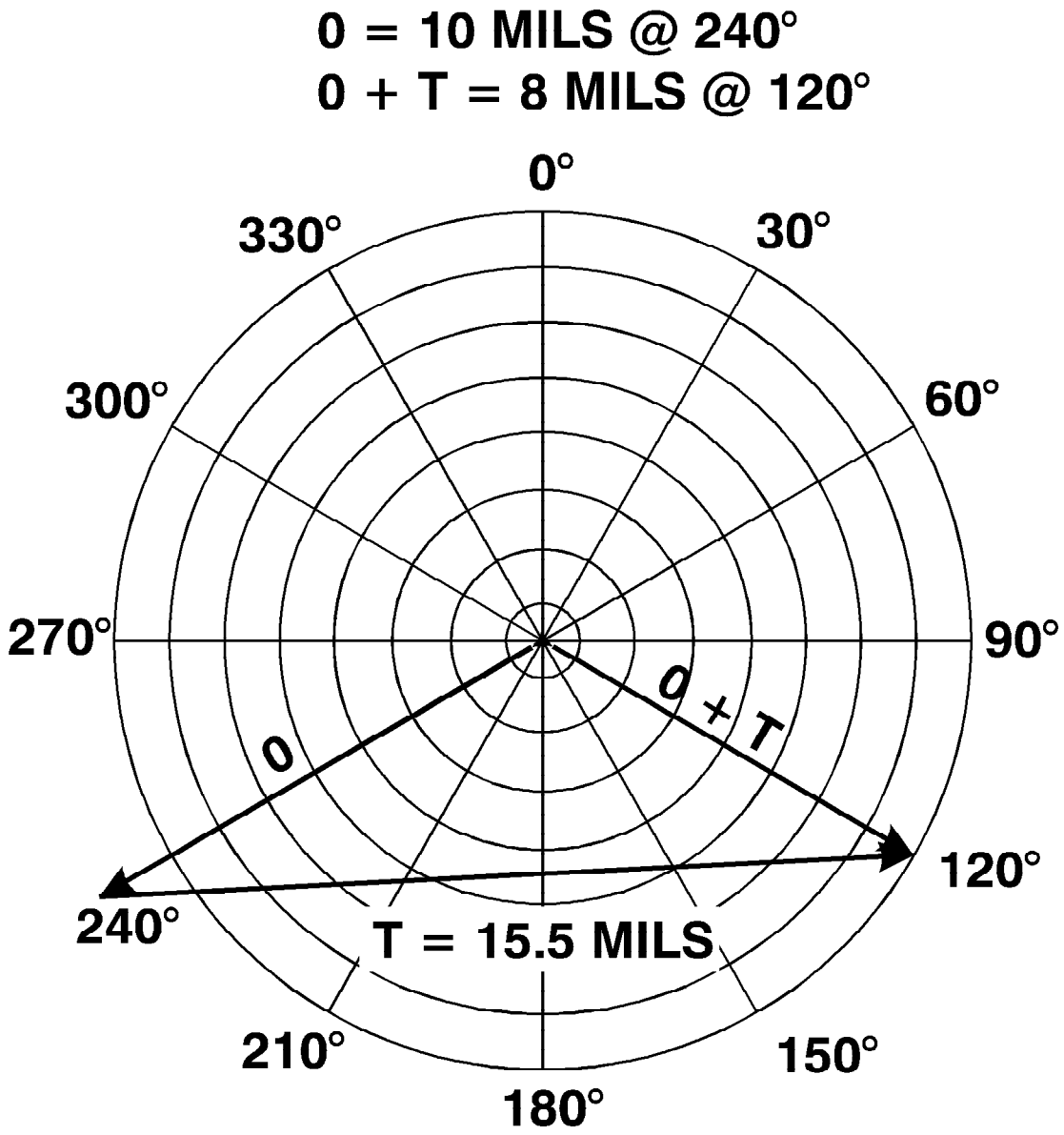

FIGURE 6.01H
STANDARD SINGLE-PLANE VECTOR SOLUTION

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TABLE 6.01A*
BALANCE QUALITY GRADES FOR VARIOUS GROUPS OF REPRESENTATIVE RIGID ROTORS IN ACCORDANCE WITH ISO 1940 AND ANSI S2.19-1975
***(Reference 20)**

Balance Quality Grades G	EW 1, 2 (mm/sec)	Rotor Types - General Examples
G 4000	4000	Crankshaft drives[3] of rigidly mounted slow marine diesel engines with uneven number of cylinders.[4]
G 1600	1600	Crankshaft drives of rigidly mounted large two-cycle engines.
G 630	630	Crankshaft drives of rigidly mounted large four-cycle engines. Crankshaft drives of elastically mounted marine diesel engines.
G 250	250	Crankshaft drives of rigidly mounted fast four-cylinder diesel engines.[4]
G 100	100	Crankshaft drives of fast diesel engines with six or more cylinders.[4] Complete engines (gasoline or diesel) for cars, trucks and locomotives.[5]
G 40	40	Car wheels, wheel rims, wheel sets, drive shafts. Crankshaft drives or elastically mounted fast four-cycle engines (gasoline or diesel) with six or more cylinders.[4] Crankshaft drives for engines of cars, trucks or locomotives.
G 16	16	Drive shafts (propeller shafts, cardan shafts) with special requirements. Parts of crushing machinery. Parts of agricultural machinery. Individual components of engines (gasoline or diesel) for cars, trucks, and locomotives. Crankshaft drives of engines with six or more cylinders under special requirements. Slurry or dredge pump impeller.
G 6.3	6.3	Parts or process plant machines. Marine main turbine gears (merchant service). Centrifuge drums. Fans. Assembly aircraft gas turbine rotors. Fly wheels. Pump impellers. Machine-tool and general machinery parts. Normal electrical armatures. Individual components of engines under special requirements.
G 2.5	2.5	Gas and steam turbines, including marine main turbines (merchant service). Rigid turbo-generator rotors. Rotors. Turbo-compressors. Machine-tool drives. Medium and large electrical armatures with special requirements. Small electrical armatures. Turbine-driven pumps.
G 1	1	Tape recorder and phonograph (gramophone) drives. Grinding machine drives. Small electrical armatures with special requirements.
G 0.4	0.4	Spindles, disks, and armatures of precision grinders. Gyroscopes.

1. The quality grade number represents the maximum permissible circular velocity of the center of gravity (CG) in mm/sec.
2. $\omega = 2\pi n/60 = n/10$, if n is measured in revolutions per minute and ω in radians per second.
3. In general, for rigid rotors with two correction planes, one-half of the recommended residual unbalance is to be taken for each plane; these values apply usually for any two arbitrarily chosen planes, but the state of unbalance may be improved upon at the bearings. For disk-shaped rotors the full recommended value holds for one plane.
4. A crankshaft-drive is an assembly which includes the crankshaft, a flywheel, clutch, pulley, vibration damper, rotating portion of connecting rod, etc.
5. For purposes of this Standard, slow diesel engines are those with a piston velocity of less than 9 mm/sec; fast diesel engines are those with a piston velocity of greater than 9 m/sec.
6. In complete engines, the rotor mass comprises the sum of all masses belonging to the crankshaft drive described in Note 3 above.

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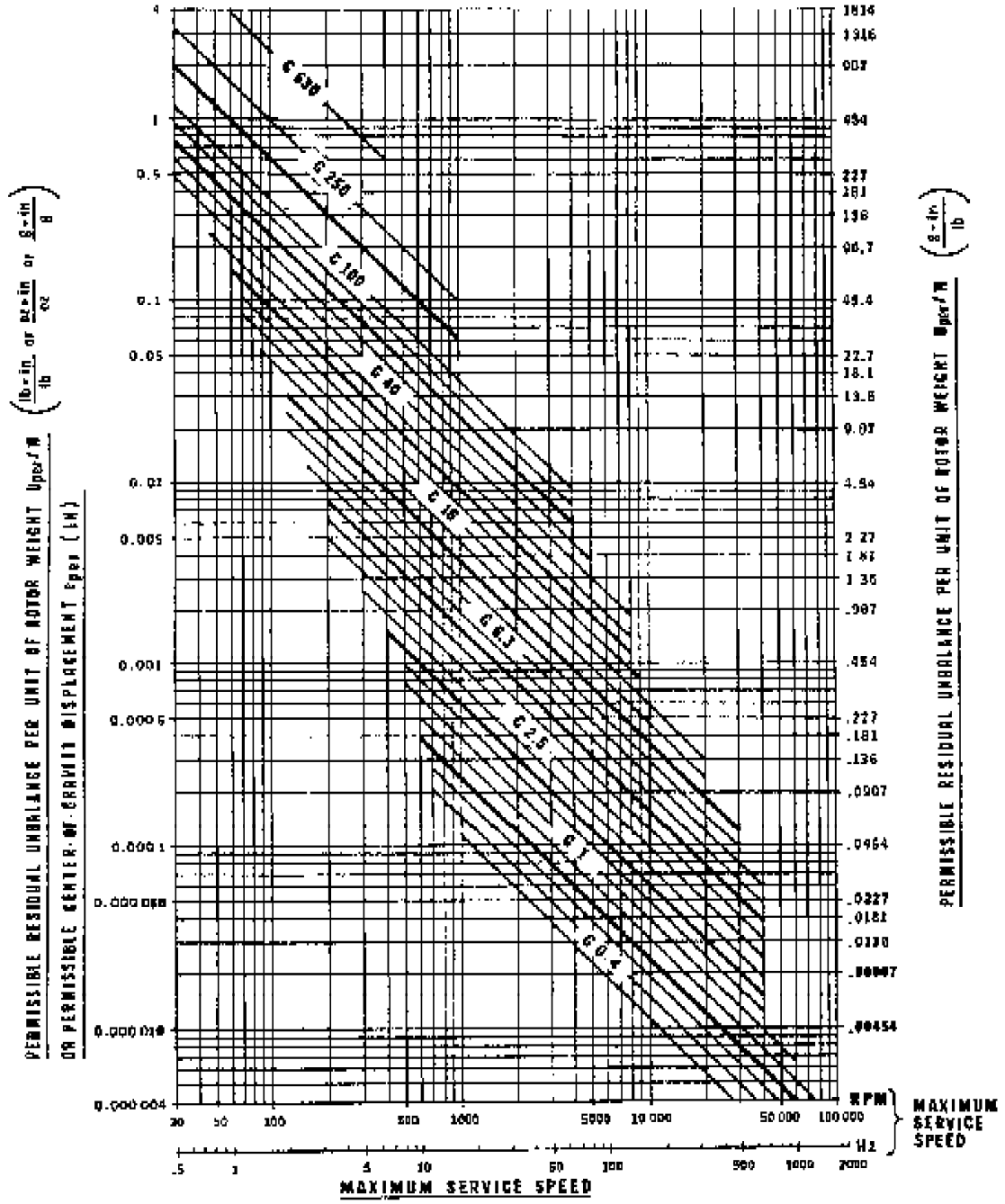
TABLE 6.01B*
MAXIMUM PERMISSIBLE RESIDUAL SPECIFIC UNBALANCE CORRESPONDING TO VARIOUS BALANCE QUALITY GRADES G, IN ACCORDANCE WITH ISO 1940

***(Reference 20)**



NOTES:

(1) In general, for rigid rotors with two correction planes, one half of the recommended residual unbalance is allowed for each plane; these values apply usually for any two arbitrarily chosen planes. (See paragraph on Applying Tolerances to Two-plane Rotors). For disc-shaped rotors the full recommended value applies to one plane.

(2) To obtain acceptable residual unbalance in oz-in per lb of rotor weight multiply lb-in value by 16.

(3) To avoid need for multiplication use appropriate Balance Tolerance Nomogram on following pages.

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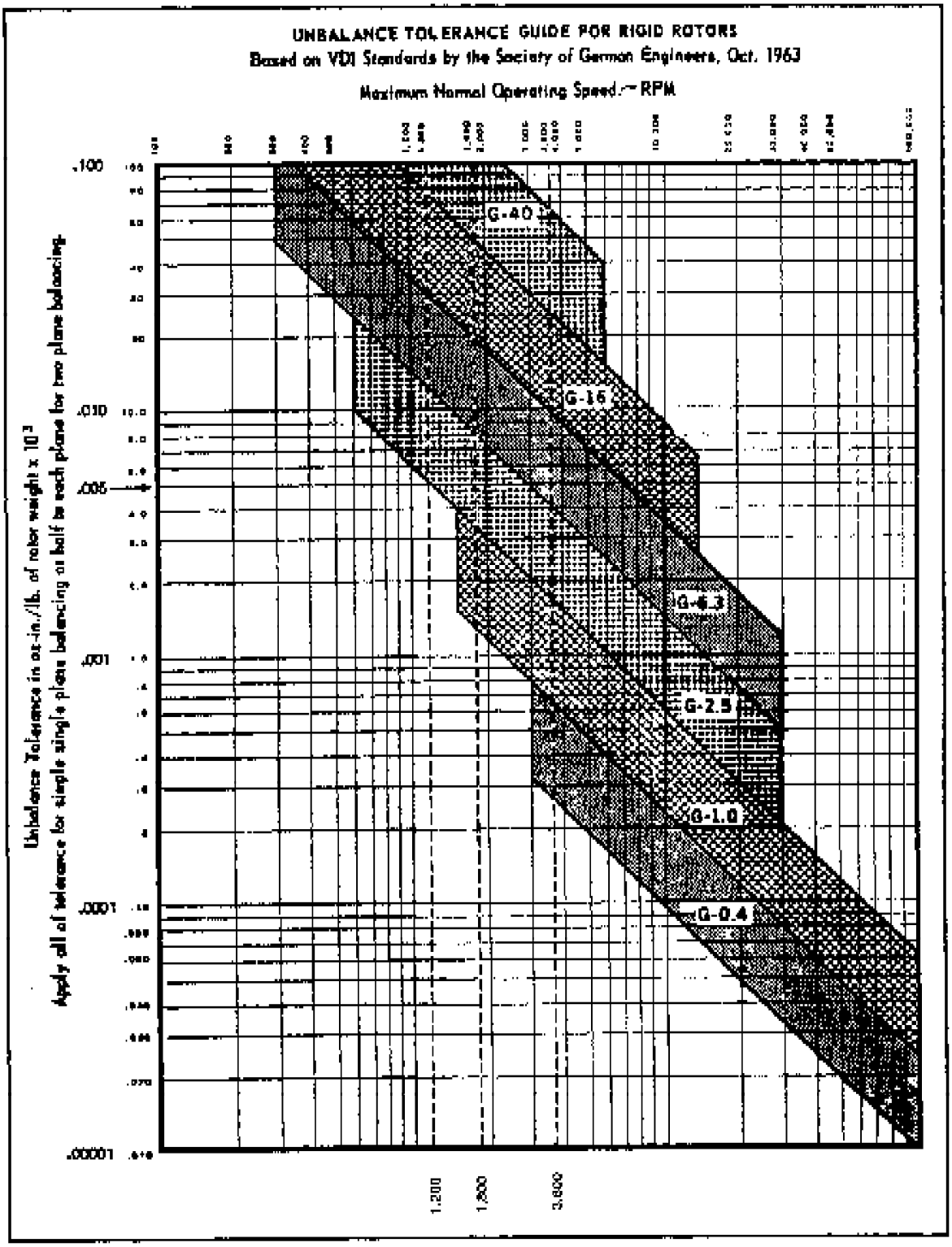
TABLE 6.01C*
EQUIVALENT ISO QUALITY GRADES SHOWN IN BANDED REGIONS
***(Reference 21)**



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6.02 ECCENTRIC ROTORS

McGraw Hill's "Dictionary of Mechanical and Design Engineering" defines Eccentricity as:

"the distance of the geometric center of a revolving body from the axis of rotation".

In other words, referring to Figures 6.02A through 6.02C, an eccentric rotor is one in which the shaft centerline does not line up with the rotor centerline. This results in more weight being on one side of the rotating centerline than the other and causes the shaft to wobble in an irregular orbit. This is inherently unstable and can be the source of troublesome vibration. Sometimes, it is possible to "balance out" part of the effect of eccentricity, but much of the displaced motion still remains. In other cases, it is not even possible to perform a good balance on rotors having more eccentricity. Today with the emphasis on higher and higher rotating speeds, it is very important that eccentricity be minimized.

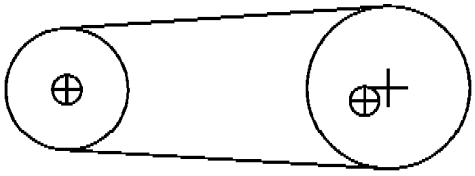

FIGURE 6.02A
ECCENTRIC SHEAVE

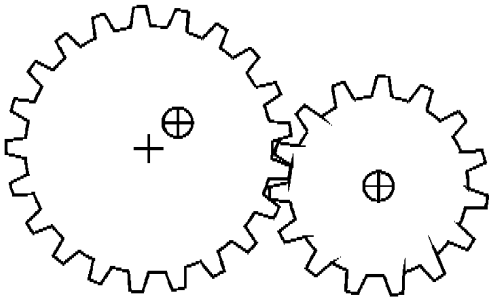

FIGURE 6.02B
ECCENTRIC GEAR

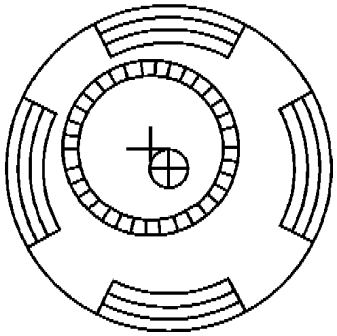

FIGURE 6.02C
ECCENTRIC MOTOR ARMATURE

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The classic spectrum of an eccentric rotor is shown on Figure 6.02D. Note that, like unbalance, the spectrum is dominated by the frequency at 1X RPM of the eccentric component, particularly when the measurement is taken in the direction through the centers of the two rotors. Looking at Figure 6.02D which shows a motor driving a fan with an eccentric pulley, note that the motor operating speed peak will be much lower than that at fan speed, particularly when taken in line with the belt direction. This eccentricity causes a very highly directional loading so that vibration at 1X RPM can be very much higher in one radial direction than in the other (depending on the amount of eccentricity).

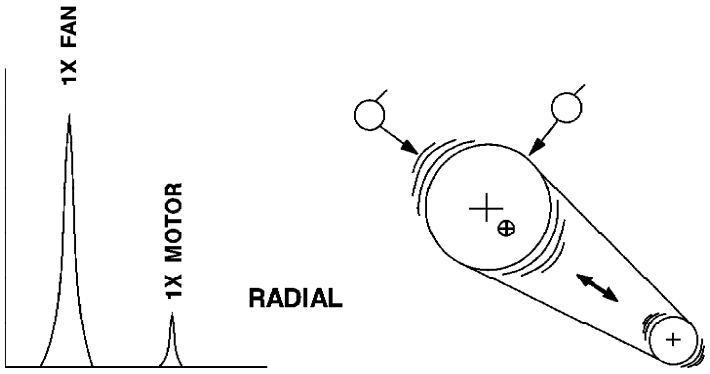

FIGURE 6.02D
TYPICAL SPECTRUM OF AN ECCENTRIC ROTOR

An eccentric rotor exhibits each of the following characteristics:

1. Some of the more common types of eccentric rotors include eccentric pulleys, gears, motor rotors and pump impellers:

    a. Figure 6.02A shows an eccentric pulley. In these units, the largest vibration most often occurs in the direction of belt tension and at the frequency of 1X RPM of the eccentric pulley. Eccentric pulleys represent one of the most troublesome sources of undesirable vibration in belt drives today. Unfortunately, the industry to date has not sufficiently policed itself to minimize eccentricity in common pulleys. Often, attempts are made to overcome pulley eccentricity after the fact by balancing. Even when this is done, balancing alone will not significantly lower the back-and-forth belt motion which results in continuous belt tension variation, depending on the position of the eccentric pulley at any instant. Plants need to protect themselves by writing eccentricity specifications into their belt drive orders if they want to maximize the life of their machinery and lower their vibration.

    b. Figure 6.02B shows an eccentric gear in which the largest vibration will occur in a direction in line with the centers of the two gears, and at a frequency of 1X RPM of the eccentric gear. The vibration signature will appear like unbalance of this gear, but it is not. If the eccentricity is significant, it can induce very high dynamic loads on gear teeth as they are forced into and out of a bind with the mating gear. Phase analysis can be used on gears having high 1X RPM vibration to differentiate whether unbalance or eccentricity is the source (See characteristic #3 below). Not only do eccentric gears result in higher 1X RPM vibration, but they also can generate high amplitude gear mesh frequencies and harmonics which will be accompanied by higher than normal amplitude sideband frequencies spaced around the gear mesh frequency at the eccentric gear RPM. Sometimes, these sidebands will be at 2X RPM of the eccentric gear. These sidebands will modulate the amplitude of gear mesh frequencies themselves.

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c. Figure 6.02C shows an eccentric motor rotor. Eccentric rotors produce a rotating variable air gap between the rotor and stator which induces pulsating vibration between 2X line frequency (7200 CPM) and its closest running speed harmonic as well as generating pole pass frequency (F_p) sidebands around 2X line frequency (see "Electrical Problem Vibration Symptoms" in Section 6.12). That is, for a 3580 RPM motor, this would be between 2X running speed and 2X line frequency, whereas for a 1780 RPM unit, it would be between 4X RPM and 7200 CPM Section 6.12 will show that an eccentric motor rotor will also generate pole pass frequency sidebands around 2X line frequency (where pole pass frequency, F_p, equals #Poles times slip frequency). Finally, the eccentric rotor motion itself will cause a variation in the magnetic field between the stator poles and rotor, thereby inducing 1X RPM vibration between the rotor and stator.

d. Eccentric pump impellers can result in unequal hydraulic forces distributed between the rotating impeller and stationary diffuser vanes. This can result not only in high vibration at pump RPM, but also at vane pass frequency and multiples (# vanes times RPM and multiples) due to a "hydraulic unbalance" induced by the eccentric impeller.

2. Attempts to balance eccentric rotors will often result in reducing vibration in one direction, but increasing it in the other radial direction.

3. Eccentric rotors may cause significantly higher vibration in one radial direction than in the other (as does resonance, wiped bearings and sometimes looseness as well). Phase analysis can be employed as an effective tool to detect whether or not the source of high vibration at 1X RPM is from eccentricity or from another 1X RPM source such as unbalance. Comparative horizontal and vertical phases usually differ by approximately 0° or 180° since the force induced by eccentricity is highly directional (rather than a 90° phase difference in horizontal and vertical as in the case of dominant unbalance problems).

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6.03 BENT SHAFT

A bowed or bent shaft can generate excessive vibration in a machine, depending on the amount and location of the bend. Like eccentric shafts, the effects can sometimes be decreased by balancing. However, more often than not, it is not possible to achieve a satisfactory balance in a shaft which has any noticeable bend. Analysts are sometimes successful in removing the bend by various techniques sometimes involving thermal treatments. In these cases, however, one must be careful not to introduce residual stresses which might later lead to shaft fatigue.

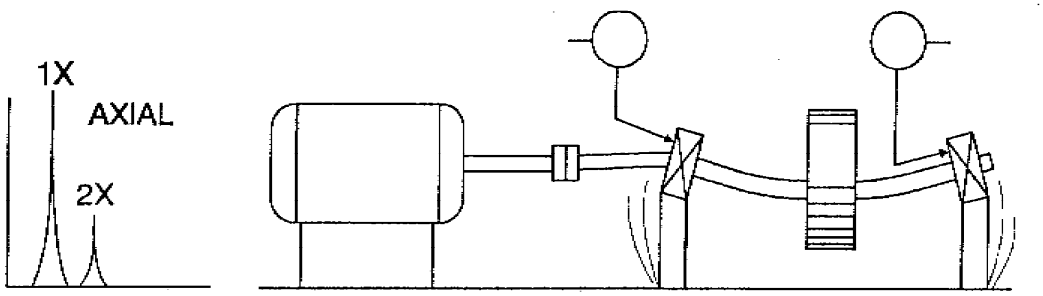

FIGURE 6.03A
BENT SHAFT SPECTRAL AND PHASE RESPONSE

Bent shafts exhibit the following characteristics:

1. Figure 1 shows that high axial vibration is generated by the rocking motion induced by the bent shaft. Dominant vibration normally is at 1X RPM if bent near the shaft center, but a higher than normal 2X RPM component can also be produced, particularly if bent near the coupling.

2. Axial phase change between two bearings on the same component (motor, fan, pump, etc.) approaches 180°, dependent on the amount of the bend (as shown in Figure 1). In addition, if one makes several measurements on the same bearing at various points in the axial direction, he will normally find that phase differences approaching 180° occur between that measured on the left and right hand side of the bearing, and also between the upper and lower sides of the same bearing.

3. Amplitudes of 1X RPM and 2X RPM will normally be steady, assuming that 2X RPM is not located close to twice line frequency (7200 CPM) which might induce a beat of the 2X RPM component with 2X line frequency if there is high electromagnetic vibration present.

4. Please note the axial phase measurements on 4 points of a bearing housing pictured in Figure 6.03B. If the shaft is bowed through or very near a bearing, you get a twisting motion by the bearing housing itself which will result in significantly different phase readings on this bearing housing in the axial direction as pictured in Drawing A of Figure 6.03B. Drawing B of this figure shows the axial phase which results from a true, straight shaft.

5. When much runout is present at the rotating mass, it appears as unbalance. When run-out at the coupling occurs, it appears as misalignment.

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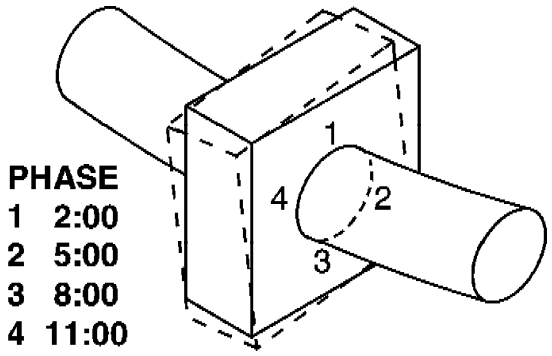
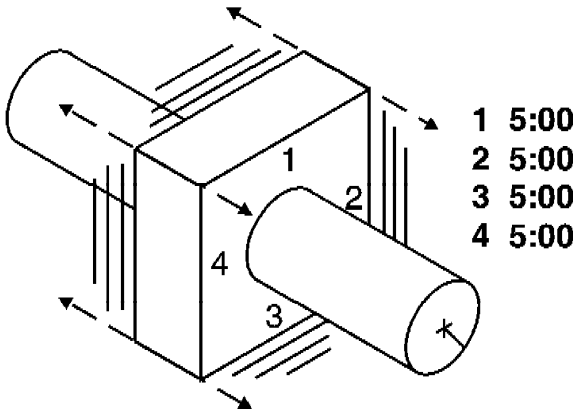


A: AXIAL PHASE MEASUREMENTS INDICATING TWISTING MOTION DUE TO A BENT SHAFT



B: AXIAL PHASE MEASUREMENTS INDICATING A TRUE SHAFT IN PROPER MOTION

FIGURE 6.03B

6. In bent shafts, amplitude can vary with the square of speed and preload. If unbalance is more of the problem than bow, vibration will decrease abruptly if operating below the first critical speed. However, if the rotor is brought above its first critical speed, unbalance amplitude will change only a small amount, whereas if the dominant problem is a bent shaft, the amplitude will again drop significantly as the speed is dropped towards the first critical speed.

7. If a rotor is located between bearings and should operate at or close to its fundamental natural frequency, it will appear to be a "bent" shaft and will display these symptoms (see Figure 6.05E in Section 6.05 on "Resonant Vibration"). However, this is only temporary. When the machine is stopped or at another non-resonant speed, it will then "straighten out".

8. When electric motors have problems such as shorted laminations, they will thermally induce a bend as the machine heats up, with the resultant vibration getting higher and higher as the rotor heats. This again will introduce bent shaft symptoms (see Figure 6.12F in Section 6.12 on "Electrical Vibration"). In this case, the shaft again will straighten when allowed to come back to room temperature if the plastic limit of the shaft material has not been exceeded. This will be covered later in the electrical problems Section 6.12.

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6.04 MISALIGNMENT

Likely the most widespread mechanical problem in industry today is misalignment. Many plants have begun to resolve a number of their unbalance problems as more and more data collectors having this capability, as well as other analyzers are acquired by plants, and balance actions are taken. However, new instruments are also now becoming available to resolve alignment problems. These include optical as well as the newer laser devices. With these newer instruments, we can now realize that machines have been operating for quite some time with much higher levels of misalignment than had previously been thought. In fact, we are learning that it is not uncommon at all to have 30% to 50% or more of machines in any plant that have high degrees of misalignment.

The trouble with such high levels of misalignment is that it induces high vibration levels leading to premature failure of expensive machine components and increased energy demands as well. Misalignment is now probably one of the leading causes of bearing failures as well.

Although vibration responds to the degree of misalignment, there is not a direct 1-for-1 relationship between the amount of misalignment (angularity and offset) and the amount of vibration. As John Mitchell states on page 182 of Reference 2:

> *"the vibration characteristics associated with misaligned flexible couplings are not a direct measure of the amount of misalignment but of the coupled system's ability to accommodate misalignment. Thus, the external symptoms of misalignment, in addition to being a function of the offset between shafts, are also affected by speed, torque, or any other condition such as corrosion or sludging which may alter the coupling's stiffness and hence its ability to accommodate a given offset."*

The first page of the Vibration Diagnostic Chart (Table 6.0) shows there are 3 types of alignment concerns including angular misalignment, parallel misalignment and a misaligned bearing cocked on a shaft. Each of these will be covered separately along with a section on coupling problems later. First, since misalignment problems are so prevalent today, a number of key facts should be considered about it including what effect it has on component lives; where it directs its potentially harmful forces; what are its spectral characteristics (harmonic content); what are its directional characteristics; what are its phase characteristics; and finally, what should be done to monitor alignment:

1. Component Failures Due To Misalignment - Misalignment can of course cause the coupling to fail, but other machine components as well. For example, if the coupling is stronger than the adjacent bearing, it can subject the bearing to excessive forces with little or no damage to the coupling. Similarly, such misalignment can detrimentally affect other components including gears, belts, sheaves, blading, etc.

2. Reaction On Free (or Outboard) End - It is possible for the highest reaction to misalignment to occur not on the bearing closest to the coupling, but on the free or outboard machine end. In these cases, incoming forces from the coupling may be strong enough to stabilize this system adjacent to the coupling and suppress the symptoms on this end.

3. Axial Vibration - Misalignment normally causes both high axial and radial vibration (as opposed to unbalance which acts mostly in the radial direction with the exception of overhung rotors).

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4. Other Sources Of High Axial Vibration - Again, while misalignment is probably the most common source of high axial vibration, there are several other sources which can generate it including:
    a. Bent Shafts;
    b. Shafts in Resonant Whirl;
    c. Misaligned Bearings Cocked on the Shaft (See Section 6.043);
    d. Resonance of Some Component in the Axial Direction;
    e. Worn Thrust Bearings;
    f. Worn Helical or Bevel Gears;
    g. A Sleeve Bearing Motor Hunting for its Magnetic Center;
    h. Couple Component of a Dynamic Unbalance.

    Therefore, when high axial vibration occurs, do not quickly jump to the conclusion that the problem is misalignment. Instead, refer especially to phase; and then to the vibration spectrum.

5. Low Axial Vibration During Misalignment - Although misalignment is classically categorized as having high axial vibration, it does not always occur. For example, the writer has experienced some cases where misalignment was the problem even though axial levels were only about 1/4 of those in the radial direction. This is quite possible for machines with predominately parallel offset versus angular misalignment.

6. Comparable Horizontal And Vertical Amplitudes - Since it has been pointed out that it is possible for a machine to have good horizontal alignment but poor vertical, it is quite possible for misaligned machines to have much higher vibration in one radial direction versus another.

7. Radial Vibration Response To Misalignment - One would think that if driver and driven shafts were horizontally offset, it would cause high horizontal forces. Although this is sometimes the case, Reference 4 states that in most cases, high horizontal amplitudes are primarily the result of vertical misalignment and vice versa.

8. 2X RPM Vibration - Often, misalignment generates a higher than normal 2X RPM vibration which can act not only in the axial direction, but also in the radial. This second operating speed harmonic is caused by asymmetric stiffness in the machine and its supports, or in the coupling. That is, there is often quite a difference in stiffness around the supporting housing, frame, foundation and coupling itself which can allow a “back-and-forth” motion with each revolution, thereby resulting in 2X RPM vibration.

9. Higher Harmonics - Misalignment can also cause large numbers of harmonics which will make the spectrum appear like looseness/excessive clearance problems. The key distinguishing feature still appears to be the high level at 2X RPM in the axial direction. Several tests have been conducted purposely misaligning units and measuring their response (Reference 3). During these referenced tests, multiple harmonics often begin to appear when the misalignment became more and more severe.

10. Phase Is Best Indicator - When high vibration occurs on a machine predominately at 1X RPM and 2X RPM, the best overall indicator of misalignment problems is phase (that is, how the machine is shaking). Phase will differentiate between a number of other potential 1X RPM and 2X RPM vibration sources. Phase behavior in response to misalignment can be summarized as follows:

    a. Probably the best indicator of misalignment problems is evaluation of phase across the coupling. Here, one is checking how the driver shaft and its coupling half is

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reacting relative to the driven shaft with its coupling half. When this phase difference across the coupling approaches 180° (±40° to 50°), misalignment is often indicated, particularly when other misalignment symptoms are present. The higher the severity of misalignment, the closer this difference will approach 180°. Similarly, the less significant are other problems such as unbalance, eccentricity, resonance, etc., the more this difference will approach 180°. Here, it is important that not only vibration spectra, but also phase measurements be captured in horizontal, vertical and axial directions on each of the bearing housings which are accessible.

b. Since it is possible for shafts to have good horizontal alignment, but poor vertical alignment (or vice versa), it is common in these cases for the horizontal phase difference to be quite different from the vertical phase difference. In fact, this is the case most of the time. In the special case where shafts have good horizontal, but poor vertical alignment, the shaft orbit itself would tend to be greatly elliptical which may make the alignment problem to appear to be eccentricity, resonance or a similar problem. That is, the amplitudes may be greatly different in one radial direction versus another. However, examination of the phase differences throughout the machine will indicate the misalignment problem.

c. When examining the phase difference on one of the rotors (just the motor, pump, fan, etc.), the radial phase differences for significant misalignment will be either 0° or 180° (±30°). This is unlike unbalance in which such phase differences could be most anything (i.e., both the horizontal and vertical phase differences might be 70°). The key here is that misalignment phase differences will approach either 0° or 180°.

d. When comparing horizontal phase differences with vertical phase differences on the same rotor, about 90% of misaligned machines will show a difference approaching 180° between the vertical and horizontal. For example, if the horizontal phase differed about 30° between the outboard and inboard bearings, the vertical phase difference would be about 210° for most misaligned rotors. An unbalanced rotor will not show this phase behavior since whatever phase difference occurs on the horizontal direction will be very close to that in the vertical direction.

In summary, phase data should always be taken if possible on machines having high vibration at 1X and 2X RPM since phase will be the key indicator in differentiating whether the dominant problem source is misalignment as opposed to other problems of similar symptoms. While other symptoms such as high axial vibration and harmonic vibration are also good symptoms, these should not receive as much weight as phase (for example, if phase does indicate misalignment, but axial vibration does not, one should give more weight to the data provided him by phase) if the vibration is high.

11. Effect Of Other Problem Sources - When other problems such as unbalance, bent shaft, resonance, etc. are present, along with misalignment, this can affect not only the vibration spectrum, but also phase behavior. For example, if both unbalance and misalignment are present, it might show high levels at both 1X RPM and 2X RPM, plus radial phase differences which may or may not approach 150° to 180°, depending on the severity of each problem (in this case, axial phase differences across the coupling will still likely approach 180°. Reference 4 suggests that when several problem sources are all present, each of them will contribute vectorially. That is, if one had polar coordinate graph paper, you might show the contribution of unbalance as a 3 mil level at 30°; the misalignment a 2 mil level at 60°; and a simultaneous eccentricity problem contributing 1 mil at 0°. The resultant vector would not show a phase at any of these 3 individual angles, but instead would produce a vector somewhere on the order of 4 mils at about 40°. This would still not be radically different in any of the original phase angles.

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However, if each of these 3 were at widely differing phase angles, the resultant phase angle and magnitude could be quite different. In this case, one would first have to solve one problem at a time (in this case, he should likely solve first the eccentricity problem; then the misalignment, and finally balance the rotor). As each problem is solved, the vibration spectra and phase will react accordingly.

12. Monitoring Alignment Change - When aligning especially critical machines, it is often helpful to monitor the alignment and see how it might change. When doing so, it is especially important to monitor phase in all 3 directions at each bearing on each machine involved in the alignment. In most machines that have thermal offsets, if the machine is brought up to speed from room temperature, it should display misalignment symptoms in the beginning which should disappear as the machine comes up to full operating temperature. For example, the phase difference across the coupling should initially be on the order of 150° to 180°, but should drop to close to 0° to 30° in the end. In addition to phase, one should monitor how the vibration spectrum changes as well as other non-vibration related variables such as bearing temperature, temperature of the support legs and oil film pressure. When monitoring alignment change, each of the following should be considered:

    a. 1X RPM - It might be better to monitor vibration at higher harmonics of 2X up to 4X RPM rather than 1X RPM since the first harmonic will be effected by so many other things (unbalance, resonance, eccentricity, bent shaft, etc.).

    b. 2X RPM - The 2X component should be a much better indicator of alignment than 1X assuming that this is not a 3600 RPM nominal motor (if so, 2X RPM will likely be very close to 2X line frequency which will "contaminate" the apparent 2X amplitude unless one is able to separate 2X RPM from 2X line frequency).

    c. 3X RPM - This 3X component may be the best indicator of alignment change if this particular machine does in fact cause an increase at 3X RPM with an alignment change. This is often the case. In such cases, it is not necessary for the 3X component to be larger than either 1X or 2X RPM, just that it be sensitive itself to alignment change.

    d. 4X RPM - The same analogy applies to 4X RPM as does 3X RPM with the exception being that this is not an 1800 RPM nominal speed machine (in which case 4X RPM would closely approach 2X line frequency at 7200 CPM).

    e. Number of Coupling Grids (or Segments) X RPM - Some coupling types include a number of grids or segments which often cause vibration at the number of grids (or segments) on one coupling half times the RPM, particularly when misalignment becomes severe. In these cases, where the coupling components themselves are effected and do respond to misalignment, this frequency will be an excellent choice to monitor since it will be well removed from any effects of unbalance, bent shaft, eccentricity or any other such source other than alignment.

6.041 Angular Misalignment:

Angular misalignment is pictured in Figure 6.04A. Each of the following characteristics are demonstrated by angular misalignment:

1. Angular misalignment primarily generates high axial vibration, particularly at 1X and 2X RPM. However, it is not unusual for either one of these peaks (1X, 2X or 3X RPM) to dominate alone.

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2. However typically, when the amplitude of either 2X RPM or 3X RPM exceeds approximately 30% to 50% of that at 1X RPM in the axial direction, angular misalignment is indicated. This assumes that there is high vibration (that is, misalignment may be of concern if 1X =.30 ips and 2X =.20 ips; but not if 1X = .03 and 2X = .02 ips).

3. Angular misalignment is best detected by 180° phase change across the coupling in the axial direction as pictured in Figure 6.04A. If each of the bearings on one of the side are moving one way, while those on the other side are moving in the opposite direction, angular alignment is highly suspect.

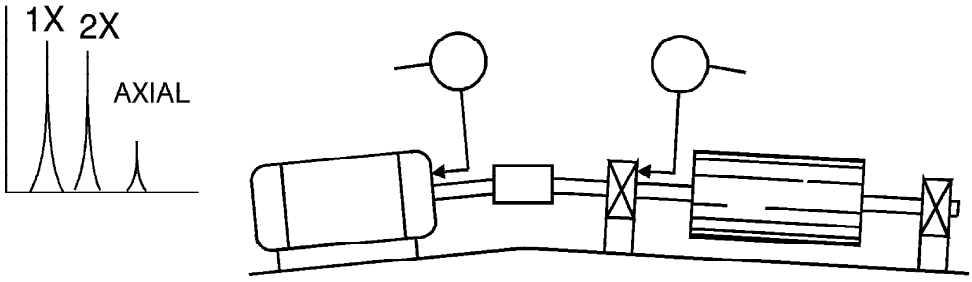

FIGURE 6.04A
ANGULAR MISALIGNMENT SPECTRAL AND PHASE RESPONSE

6.042 Parallel Misalignment (Also known as Radial Offset Misalignment):

Radial misalignment is pictured in Figure 6.04B. It displays each of the following characteristics:

1. Parallel misalignment primarily affects radial vibration as opposed to angular which affects axial.

2. Like angular alignment problems, parallel misalignment causes phase to approach 180° difference across the coupling, but in the radial direction (horizontal or vertical).

3. Radial misalignment is often indicated in a spectrum when 2X RPM exceeds approximately 50% of the amplitude at 1X RPM, but its height relative to 1X RPM is often dictated by the coupling type and construction. It is not uncommon for 2X RPM to exceed that at 1X RPM, particularly when the parallel misalignment becomes severe.

4. When either angular or parallel misalignment becomes severe, each can generate an array of harmonics ranging up to and including the 4th through the 8th harmonic. In this case, the severe misalignment spectrum can appear to be mechanical looseness (see page 1 of Table 6.0 Diagnostics Chart).

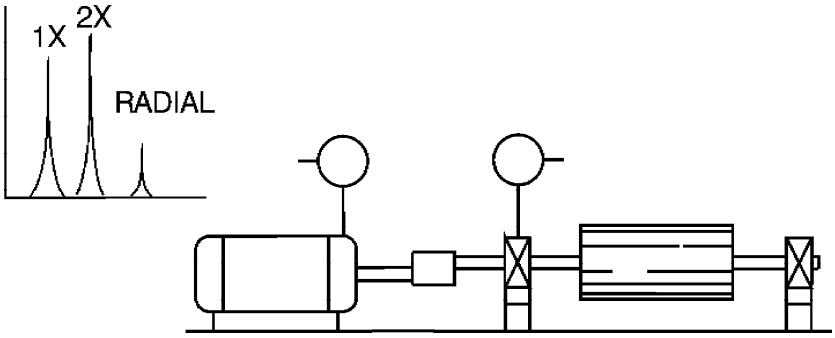

FIGURE 6.04B
PARALLEL MISALIGNMENT SPECTRAL AND PHASE RESPONSE

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6.043 Misaligned Bearing Cocked on the Shaft:

When either a sleeve or rolling element bearing is misaligned or cocked on the shaft, it can cause high vibration and unusual loading. If it is detected, it should quickly be resolved before causing premature component failures. This problem is pictured in Figure 6.04C. Each of the following characteristics are indicative of a misaligned bearing on a shaft:

1. A cocked bearing will normally generate considerable axial vibration which can affect not only that at 1X RPM, but also 2X RPM as well.

2. If phase is measured in the axial direction at each of 4 points 90° apart from each other as shown in Figure 6.04D, a cocked bearing will be indicated by a 180° phase shift from top to bottom or from side to side.

3. Attempts to align the coupling or balance the rotor will not alleviate the problem. The effected bearing must be removed and correctly installed.

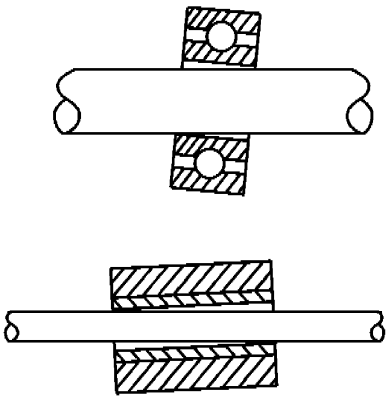

FIGURE 6.04C
MISALIGNED BEARINGS COCKED ON SHAFT

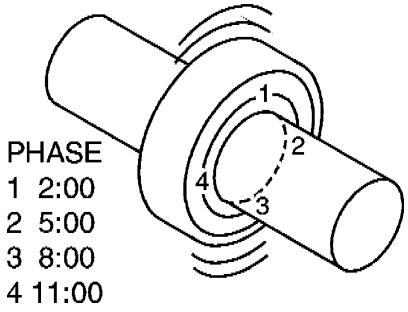

FIGURE 6.04D
AXIAL PHASE MEASUREMENTS INDICATING A COCKED BEARING ON A SHAFT

6.044 Coupling Problems:

It is often difficult to tell from vibration signatures or phase analysis whether the problem is misalignment or a coupling problem. Each of the many types of couplings has a different effect on the response of the machinery to which it is coupled. Other factors affecting its response include spacing between shafts, shaft diameter and bearing type. However, problem couplings do display the following characteristics:

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1. 3X RPM will often respond to a coupling problem, particularly for a coupling having a spacer that is too long or too short. In these cases, the radial spectrum will normally indicate a fairly noticeable 3X running speed vibration, and that in the axial direction will show a much higher 3X RPM component. These problems are resolved by either correctly sizing the spacer or repositioning either the driver or driven equipment.

2. Gear Type Couplings can experience coupling lockup where the frictional force developed at gear teeth is greater than the applied force causing the coupling to become a rigid member. Friction welding of teeth can occur at this point, particularly if there is a lack of lubrication. A locked coupling can cause severe problems and may lead to thrust bearing failure if it results in the thrust load of 2 machines being applied to only one thrust bearing. Also, if the teeth do weld together and then break loose, it leaves pit marks on the coupling teeth. Coupling lockup can be broken temporarily either due to a change in load or by striking the coupling with a mallet or a piece of wood. However, this coupling should be closely inspected as soon as possible looking for tooth damage, lubrication problems and alignment problems, replacing if necessary.

   Coupling lockup will normally cause an increase in both axial and radial vibration with axial vibration normally being higher. Most of the time, the 1X RPM is most effected. However, certain types of couplings will generate a frequency distribution resembling a "Christmas Tree" effect. In these cases, many harmonics can appear with the vibration dropping approximately 25% from one harmonic to the next (Reference 5). What gives the spectrum a "Christmas Tree" effect is that there is a fairly uniform drop of about 25% all the way from the 2nd through the 5th or 6th harmonic.

3. A loose coupling is likely to cause sidebands around blade pass frequencies (#blades X RPM) and mesh frequencies (#teeth X RPM) as shown in Figure 6.04E (however, side-banding of blade pass and mesh frequencies does not always indicate a loose coupling). This is caused by the fact that a loose coupling does not drive the rotating equipment at a uniform speed, but rather, pulses at multiples of the shaft speed causing its running speed to modulate these other frequencies. Therefore, a signal similar to that shown in Figure 6.04E with equally spaced sidebands at coupling RPM can mean the coupling is loose (either from a poor fit on the shaft or from worn coupling components).

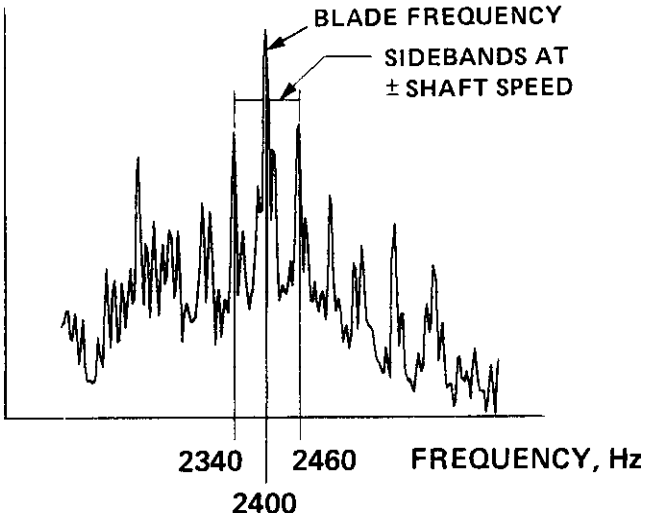

FIGURE 6.04E
LOOSENESS OF COUPLING INDICATED BY COUPLING RUNNING SPEED SIDEBANDS ABOUT BLADE PASS FREQUENCY

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6.05 MACHINERY FAILURES DUE TO RESONANT VIBRATION

Resonant vibration problems are much more commonplace on today's machinery than it generally has been assumed. As a result, machinery components are often subjected to excessive vibration which often leads to catastrophic failure or, as a minimum, premature failure long before the parts should have failed. Therefore, it is important that industry learn both how to detect resonance and how to solve resonant vibration problems.

Resonance occurs when a forcing frequency coincides with a natural frequency. This may be a natural frequency of the rotor, but often can be a natural frequency of the support frame, foundation, or even drive belts. Forcing frequencies include those from sources such as unbalance, misalignment, looseness, bearing defects, gear defects, belt wear, etc.

Figure 6.05A helps illustrate resonance. It shows a graph of magnification factor (Q) on the vertical axis versus frequency ratio (f/f_n) on the horizontal axis. The magnification factor, Q, is actually an amplitude ratio which relates how much vibration is amplified when a machine passes through resonance. As the drawing on Figure 6.05A shows, the magnification factor is a ratio of dynamic deflection at a forcing frequency (X_o) to static deflection due to load (X_{ST}). In other words, it compares shaft deflection at rest versus that with it rotating. The frequency ratio is the ratio of forcing frequency (f) to natural frequency (f_n). The upper graph in Figure 6.05A shows that when the frequency ratio approaches 1.0 (or when the forcing frequency "tunes in" to a natural frequency), the vibration amplitude can become extremely high, depending only on the amount of damping in the system. In fact, the equation below the graph shows that the only thing which limits an infinite vibration amplitude when at resonance is the system damping (where damping is ζ, also known as the damping factor). In common machine structures made of steel, aluminum, cast iron and so forth, the damping factor is normally 0.05 or less, which shows that resonant amplification will be 10X for this damping factor, and can even approach amplifications of 50X for systems with lower damping (see upper graph in Figure 6.05A). Therefore, such vibration can easily lead to premature, or even catastrophic machine failure.

Please note the lower graph in Figure 6.05A plotting Phase Lag versus frequency ratio for various levels of damping. Note that when a machine passes through resonance, it will see a phase change of 90° when right on the natural frequency, and phase will continue to rapidly change when still in the vicinity of the natural frequency. Finally, phase will change almost 180° when completely through the resonance. The total phase change and the rate at which phase will change for given distances away from the natural frequency will be governed by the amount of damping which can be seen in Figure 6.05A. The lower the damping, the greater will be the rate of phase change.

Figure 6.05B shows a diagram of something known as a Bode' Plot. Here, both vibration amplitude and phase are plotted on the vertical axis versus shaft speed (RPM) on the horizontal, as a machine is either brought from rest to full operating speed or when it is shut down and allowed to coast to a stop. Looking at the amplitude versus RPM curve, note that this particular machine went through two resonant events as it coasted down from approximately 6500 RPM. First, the amplitude increased from only .125 mil to about .872 mil when it reached about 4850 RPM; then it quickly dropped back down to about .130 mil again. Then, it excited another natural frequency at about 2450 RPM when levels increased up to about .600 mil. Finally, it dropped again and continued to decrease in amplitude until it came to rest. This is a typical Bode Plot for common rotating machinery. In this example, this machine would run well if it were operated from approximately 0 to 2200 RPM; or from 2700 to 4300 RPM; or from 5300 to 6500 RPM. However, it would have serious resonant amplification if it were operated either from about 2300 to 2600 or from 4600 to 5100 RPM.

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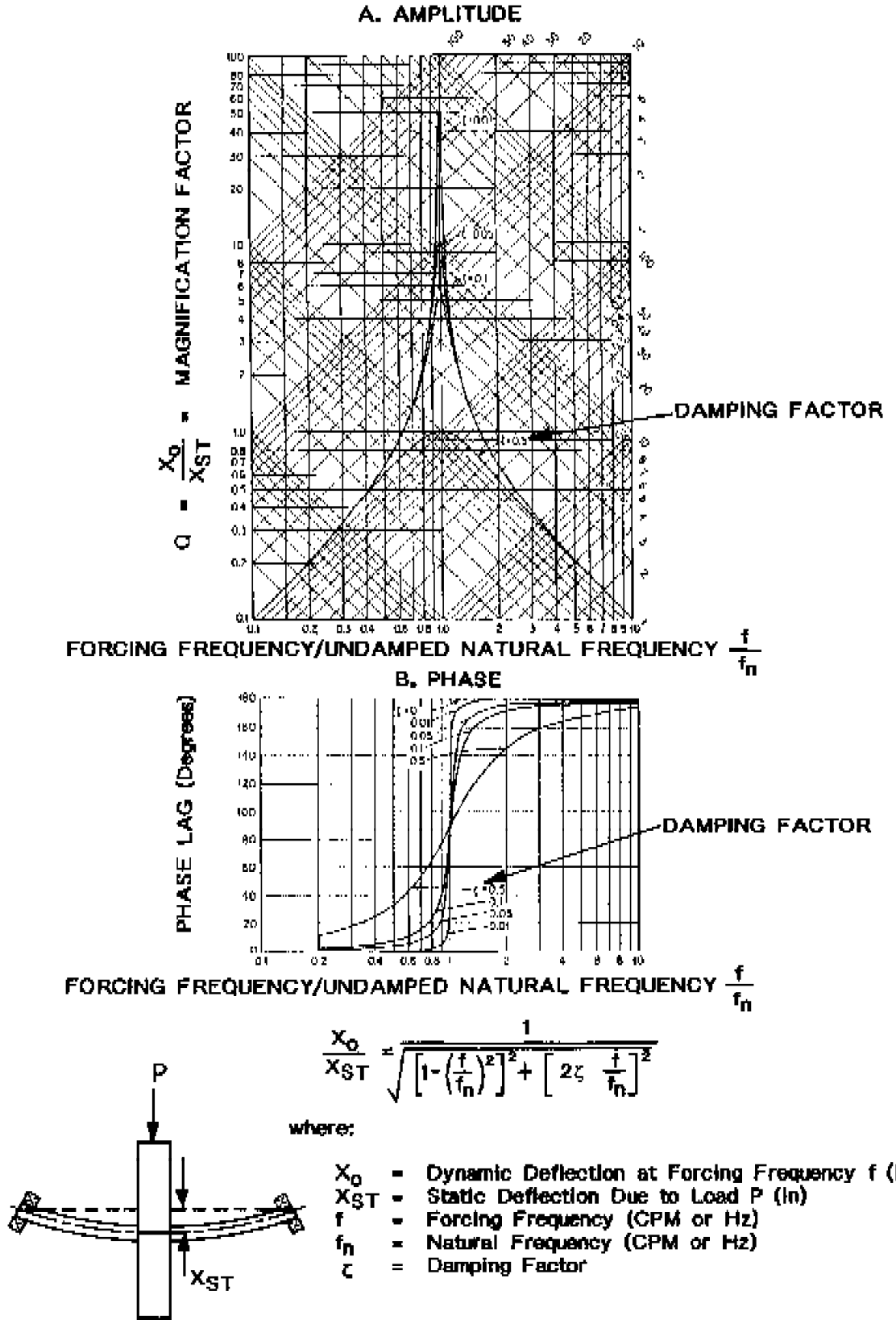
A. AMPLITUDE
Q = $\frac{X_o}{X_{ST}}$ = MAGNIFICATION FACTOR
DAMPING FACTOR
FORCING FREQUENCY/UNDAMPED NATURAL FREQUENCY $\frac{f}{f_n}$
B. PHASE
PHASE LAG (Degrees)
DAMPING FACTOR
FORCING FREQUENCY/UNDAMPED NATURAL FREQUENCY $\frac{f}{f_n}$
$\frac{X_o}{X_{ST}} = \frac{1}{\sqrt{\left[1-\left(\frac{f}{f_n}\right)^2\right]^2 + \left[2\zeta \frac{f}{f_n}\right]^2}}$
P
X_{ST}
where:
X_o = Dynamic Deflection at Forcing Frequency f (In)
X_{ST} = Static Deflection Due to Load P (In)
f = Forcing Frequency (CPM or Hz)
f_n = Natural Frequency (CPM or Hz)
ζ = Damping Factor

FIGURE 6.05A
RESONANT AMPLIFICATION CURVE AND PHASE CHANGE AT RESONANCE

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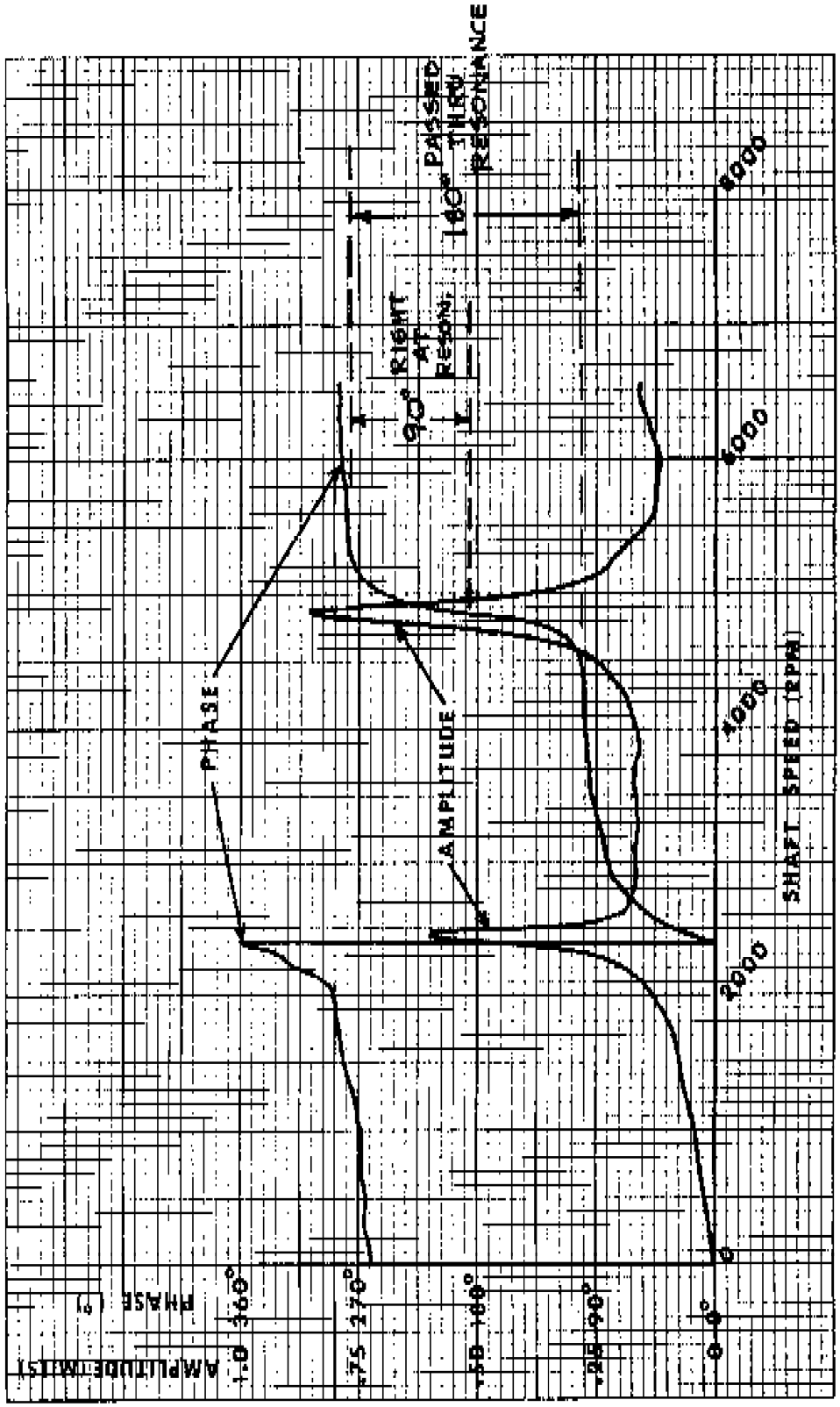

FIGURE 6.05B
BODE' PLOT - VIBRATION AMPLITUDE AND PHASE VERSUS RPM

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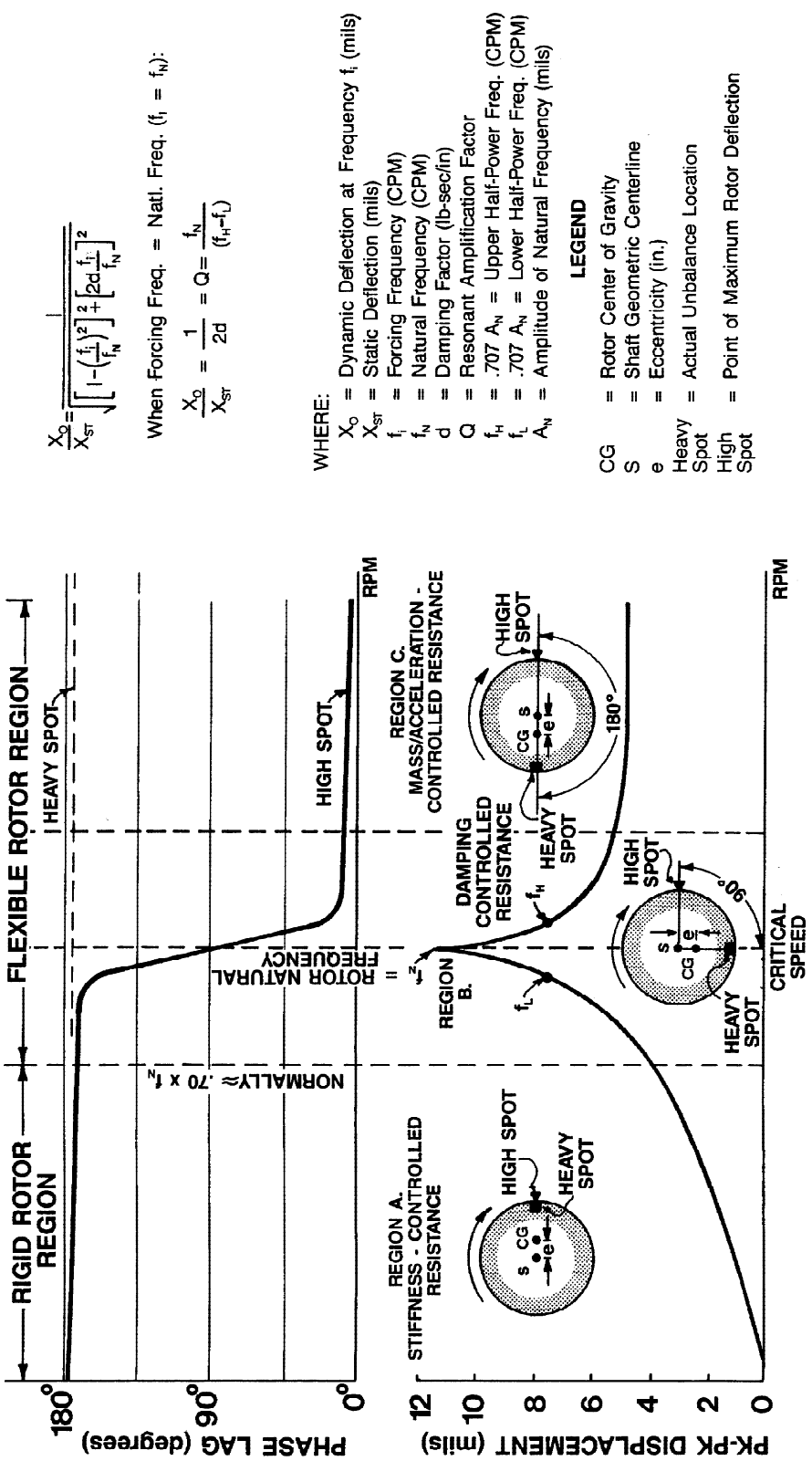

FIGURE 6.05C
CHANGE OF VIBRATION DISPLACEMENT AND PHASE LAG WITH RPM ABOVE, BELOW, AND AT ROTOR RESONANCE (BODE' PLOTS)

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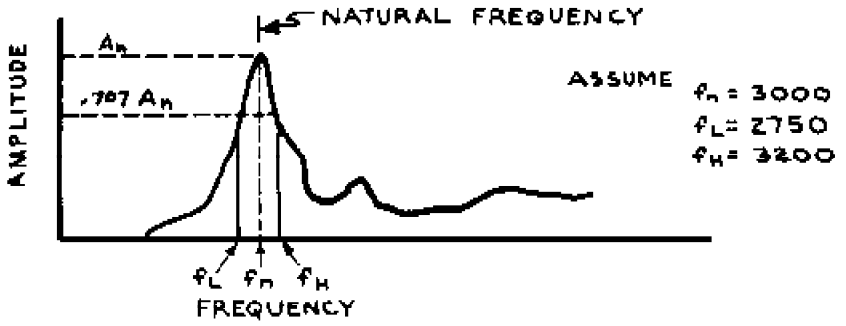


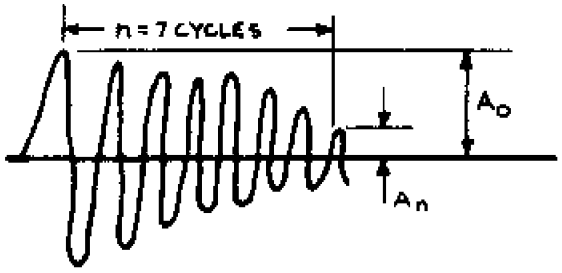
HALF POWER POINTS AT (.707)(A_n) DEFINE f_L AND f_H

$$\frac{X}{X_0} = Q = \frac{f_n}{f_H - f_L} = \frac{3000}{3200-2750} = 6.67$$

$$Q = \frac{1}{\sqrt{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left(2\frac{c}{c_c}\frac{\omega}{\omega_n}\right)^2}} \quad \text{AT CRITICAL} = \frac{1}{2\frac{c}{c_c}\frac{\omega}{\omega_n}}$$

$$6.67 = \frac{1}{2\frac{c}{c_L}} \qquad \frac{c}{c_c} = \frac{1}{(2)(6.67)} = .075$$

DAMPING FROM AN IMPULSE TEST :



LOG DECREMENT $\delta = \frac{1}{n} Ln\left(\frac{A_0}{A_n}\right)$ ← KNOWN FROM IMPULSE TEST

$$\frac{c}{c_c} = \frac{\delta}{\sqrt{4\pi^2 + \delta^2}}$$

FIGURE 6.05D
CALCULATING DAMPING FROM A COASTDOWN AND IMPULSE TEST

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Please note one other important fact shown in Figure 6.05B. When this machine passed through resonance, the vibration phase changed exactly 90° right at the natural frequency as predicted and continued to change almost 180° until it suffered no more resonant amplification. Note that this happened at both resonant events in this machine example. The second event might be a little confusing, but note that the phase changed from about 90° to 0° at resonance and then continued from there to about 270° when it passed out of trouble (360° phase is the same as 0°).

Figure 6.05C is likewise a Bode' Plot and shows how a rotor responds in each of three regions (A, B and C), each of which show a unique relationship of the rotor heavy spot (actual unbalance location) relative to its high spot (point of maximum rotor deflection). In region A, where stiffness controls dynamic resistance, there is little phase difference between the heavy spot and high spot. That is, if it were possible to measure rotor displacement with a dial indicator in region A, the greatest displacement would occur almost instantly at the heavy spot pass by the dial indicator during each shaft revolution. Also in region A, unbalance vibration response will increase with the square of the speed (tripling the speed will result in about 9 times higher displacement).

In Figure 6.05C, as the rotor increases speed and enters region B, damping causes the high spot to begin to lag the heavy spot of the shaft by a predictable and repeatable angular amount. This angular lag (phase) is caused by delay in the time it takes for the heavy spot to create shaft displacement. In fact, as the rotor approaches the first balance critical speed, Figure 6.05C shows that the maximum displacement (high spot) does not occur until the heavy spot is already passed by one-fourth of a revolution (or 90°). When this 90° phase lag occurs at resonance, the only parameter limiting infinite vibration response is the amount of damping (z).

Figure 6.05C then shows what happens when the rotor begins to pass through the amplitude magnification region and out of resonance (into region C). Phase lag continues to change dramatically, finally approaching 180° when it has passed completely through the critical. Now, what this means is that the high spot (point of maximum rotor deflection) now will actually lag the heavy spot a full 180°; that is, the heavy spot will complete one-half a revolution before the maximum rotor deflection even occurs. In this region C, the dynamic resistance is largely controlled by mass with almost no resistance applied from either spring stiffness or damping as the rotor system responds almost as a pure mass. Also interesting is the fact that in region C well beyond the first critical (and well before the vicinity of the second critical), the displacement becomes nearly constant even though centrifugal force (F_C) continues to increase with the square of speed. The reason for this is that the resistance to vibration in region C is governed by the product of mass times acceleration ($m w^2$) where $w = 2 p f/60$. Therefore, even though centrifugal force increases with the square of speed, dynamic resistance to motion likewise increases with the square of speed counteracting centrifugal force, and resulting in a displacement which is near constant.

Figure 6.05D illustrates how far away one must move a natural frequency to avoid resonant amplification. Here, a figure is shown graphing vibration amplitude versus frequency. When at resonance, this machine has amplitude A_n at frequency f_n. Resonant amplification occurs at so-called half-power points which are defined as points below and above natural frequency at 70.7% of the amplitude at resonance (see f_L and f_H). The equation below the diagram shows how the magnification factor (Q) can be calculated if both the natural frequency and half-power points are known. In the example given, resonant amplification is 6.67; that is, resonance amplified vibration by a factor of almost 7X. Then, a calculation of damping factor was made by solving the equation given in Figure 6.05A. This results in a damping factor of about .075 which is a little higher than many common machine materials.

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Finally, Figure 6.05D also shows how damping can be calculated from an impulse test when something such as a rubber-coated hammer is used to impact a machine to excite system natural frequencies. The time waveform in Figure 6.05D shows the response of the machine from the time of impact until damping can dissipate the vibration. Here, one would look at the time waveform and take the number of cycles (n) that best show the system response. In the example shown, A_o is the amplitude at impact where amplitude A_n is the amplitude at the nth cycle.

Therefore, it is most important that plants recognize a resonance when it occurs and make immediate attempts to resolve the problem rather than continuing to subject their expensive machinery to excessive vibration unnecessarily. With the instrumentation available today, there is no reason why this cannot easily be done, even by rather inexperienced vibration personnel. If this is accomplished, these machines will have significantly enhanced lifetimes and, during this life, will have significantly lower costs associated with required repairs.

6.051 Identifying Characteristics of Natural Frequencies that Help "Give Them Away":

Resonant vibration is indicated by the following characteristics:

1. Lack of Response to Balance Attempts:

Examination of Figure 6.05C explains why it is almost impossible to balance a machine which is at or near resonance. The machine will show a dramatic phase change for even a small change of speed in the vicinity of resonance which can approach 180°. Thus, to balance this rotor, it will be necessary either to change the natural frequency, operate it at a different speed, or remove the rotor from the machine and balance it on a fixed balancing stand.

2. Highly Directional Vibration:

Resonant vibration will cause much higher vibration in one direction as compared to the other two triaxial directions (for example, horizontal might be 10X higher than either vertical or axial levels). If resonant, it is common for vibration in the resonant direction to be 5 to 15X higher than that in the other two triaxial directions. This fact is now used in many expert diagnostic software systems when looking for a possible resonance. This is also why it is important that readings be taken in <u>all 3 directions</u> at each bearing during routine Predictive Maintenance Surveys.

3. Phase Behavior in the Resonant Measuring Direction:

Resonant frequencies will show a great change of phase with RPM in the direction in which the machine is resonant since phase will change 90° right at a natural frequency, and almost 180° when passing completely through resonance, depending on the amount of damping present. On the other hand, phase changes in non-resonant measurement directions may be small at the same time since they are not undergoing a natural frequency event.

4. Probable Phase Difference in Measuring Direction Perpendicular to Resonant Measuring Direction:

If one radial direction is resonant, the phase difference when the transducer is moved 90° to measure in the other radial direction will likely approach either 0° or 180°, depending on which side of the bearing that the pickup is placed (instead of about a 90° difference as in the case of dominant unbalance). That is, if the horizontal direction is resonant, the horizontal phase will likely either equal the vertical or differ from it by almost 180°. This

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is due to the fact that an additional 90° phase change is introduced by operating at a natural frequency. In either case, either a 0° or 180° horizontal-vertical phase difference represents highly directional motion characteristic of resonance (or an eccentricity).

5. Characteristic Shape of a Resonant Peak:

Normally, a resonant peak will have a rather wide "apron" at its base as opposed to the much narrower aprons for non-resonant peaks. That is, the base of a resonant peak will usually be wider than those of surrounding non-resonant peaks.

6. Frequencies at Which Resonances Can Occur:

Resonance does not occur only at 1X RPM. It can respond to any forcing frequency that happens to coincide with a natural frequency. In these cases, it is meaningful to compare the amplitude of this frequency in this direction with the same frequency in the other two triaxial directions. If resonant, the frequency should be much higher in one of these three directions. This frequency might be a peak at 4X, 5X or 6X RPM (or even higher) corresponding to a blade pass frequency (BPF), bearing defect frequency, gear mesh frequency (GMF), or even to a mechanical looseness condition. If action is taken on the source of this exciting frequency resulting in lowering of the amplitude of the forcing frequency itself, it may also lower the response of this natural frequency to the forcing frequency. Remember that Resonant Vibration = Forcing Frequency Vibration (f) X Magnification Factor (Q).

7. Excessive Motion and Dynamic Stress of any Resonant Member:

Not only must one be concerned about resonance of the machine rotor (rotating assembly), but also with exciting natural frequencies of the support frame, foundation and even connected piping and/or duct work. Very often, fatigue failures will occur in the connected frame and/or piping due to the fact that they happen to be resonant to the forcing frequency coming from the machine. Resolution will require either lowering the amplitude source of the forcing frequency in the machine, isolating the resonant frame member from the machine, changing the rotor speed, or changing the natural frequency of the frame member itself.

8. Resonance Can Suddenly Occur on a Long-Running Machine Never Before Subjected to It:

Resonance can suddenly occur without warning in a machine running for years with little or no problems. For example, bearing wear can reduce stiffness of a shaft/bearing system lowering the natural frequency which might put it in resonance with a forcing frequency. Also, simple journal bearing replacement can cause a change in natural frequency which could put the rotor in resonance if the bearings are not properly blued and scraped to establish good, continuous contact with the shaft. After this has been done, one can simply resolve the problem after installing the bearing and checking for required clearance specifications and proper bearing alignment. The reason that each of these items affect resonance is that they each can have a direct effect on stiffness.

9. Possible Effect of Coupling Changes:

Coupling changes can sometimes result in a different torsional natural frequency, particularly if a different diameter coupling is used, or a different coupling type or construction is employed. This can introduce significant vibration in bearings on either side of the coupling.

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10. Summary of Actions Needed to Resolve Resonance Problems:

a. Change Stiffness:

Resonance can sometimes be resolved by increasing the stiffness of pedestals or frames, clamping of feet, removing distortion induced by shimming, or using reduced clearance bearings (which will affect stiffness, and therefore natural frequency).

b. Adding or Removing Weight (Mass):

Although it is normally simpler and less expensive to change stiffness, natural frequency can likewise be changed by adding or removing weight. However, it is important to point out this effect will only occur on the part that is resonant.

c. Adding Isolation Mechanisms:

Vibration isolation can be effective on a machine which is forced into resonance at one of its natural frequencies due to an incoming vibration from external equipment. For example, a properly designed pump operating at 1780 RPM and having a 750 CPM natural frequency may have a problem if a large fan operating nearby at about 750 RPM is out of balance. In this case, it would be necessary to either isolate the fan from the pump, balance the fan, or isolate the pump itself from the floor.

d. Adding Damping Materials:

Resonant amplitude amplification can be lowered by increasing damping of the part that is resonant (however, it is important to point out that damping itself will be of little help if the part on which the damping materials are placed is not resonant). Damping is the only parameter that prevents catastrophic failure when a part is resonant.

e. Installing Dynamic Vibration Absorber:

A "tuned" dynamic vibration absorber can sometimes be installed on equipment on which other reduction techniques either have not proven practical, or have not sufficiently reduced vibration to acceptable levels. However, this device is only effective in those situations where only one dominant frequency is causing excessive vibration and this frequency is a natural frequency of the part on which the tuned damper is to be mounted. In these cases, the tuned damper can be designed with the natural frequency equal to the troublesome frequency and will respond 180° out of phase with the resonant motion which will in effect “cancel out” the problem natural frequency.

f. Precision Dynamic Balancing:

Since resonant vibration equals forcing frequency vibration X magnification factor, if one cannot effectively move the natural frequency out of resonance or lower the magnification (by addition of damping materials), one can still lower resonant vibration by reducing the sources of the forcing frequencies (for example, unbalance). In this case, if the balancing amplitude can be significantly reduced by several orders, the result can be acceptable vibration, even though at resonance. However, it is important to point out that it may be necessary to remove the rotor in order to even attempt to balance it because of the tremendous phase changes at resonance previously mentioned.

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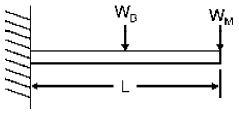
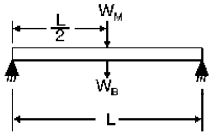
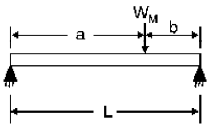
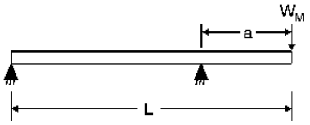
6.052 How Natural Frequencies Can Be Approximated For Overhung Rotors and Machines with Loads Supported Between Bearings:

Following below is Equation 6.05B which applies to calculation of natural frequency for a single degree-of-freedom system like those shown in Table 1:

$$f_n = \frac{60}{2\pi}\sqrt{\frac{K}{m}} = \frac{60}{2\pi}\sqrt{\frac{g_c}{\Delta}} \qquad \text{(Eqn. 6.05B)}$$

where: f_n = Natural Frequency (CPM)
g_c = Standard Gravitational Constant
= 32.2 ft/sec^2 = 386 in/sec^2 (SEA LEVEL)
K = Stiffness (lb/in)
m = Mass (lb_f - sec^2/in) = Weight/g_c
Δ = Deflection (in)

Table 6.05A provides formulas for approximating natural frequency for cantilevered, simply supported and overhung rotors. In the simply supported case, the applied load (W_m) can act either at the center of this shaft span (Case B) or at any point along the shaft between the bearings (Case C).

CASE	DIAGRAM	FORMULA FOR f_n
A.		$f_n = 324.96\sqrt{\frac{EI}{L^3(W_M + 0.24\,W_B)}}$
B.		$f_n = 1302\sqrt{\frac{EI}{L^3(W_M + 0.49\,W_B)}}$
C.		$f_n = 324.96\sqrt{\frac{EIL}{a^2 b^2 W_M}}$
D.		$f_n = 324.96\sqrt{\frac{EI}{a^2 L W_M}}$

f_n = Natural Frequency (CPM)
E = Modulus of Elasticity of Beam or Shaft Material (lb/in^2)
I = Principal Moment of Inertia of Beam Section (in^4)
W_M = Applied Load (lb)
W_B = Weight of Beam or Shaft (lb)
L = Length of Beam or Shaft (in)
a, b = Dimensions as Shown (in)

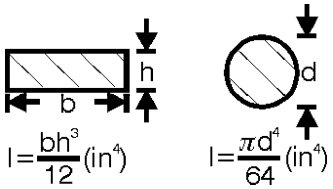


TABLE 6.05A
NATURAL FREQUENCY FORMULAS FOR OVERHUNG ROTORS, CANTILEVER BEAMS AND SHAFTS MOUNTED BETWEEN BEARINGS

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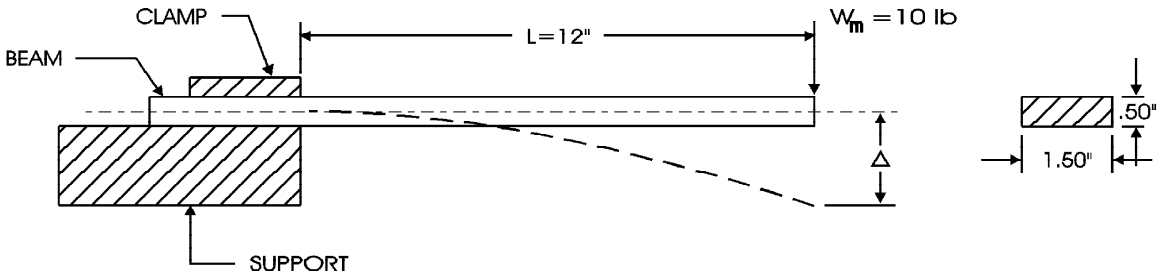
In teaching seminars, the author has found that there is often a lack of understanding if formulas and tables are simply presented without examples. Therefore, several examples applying the above equations and showing how they can be used to calculate natural frequency will now be provided:

Example 1 - Applying Cantilever Beam Natural Frequency Formulas:



Given: Rectangular Steel Beam (E = 29,600,000 lb/in^2), 12 inch cantilevered portion, 1.50" wide by .50" deep; Applied Load (W_m) of 10 lb at End.

Part A - What is system natural frequency if you neglect the beam weight?

Moment of Inertia I, for a rectangular beam,

$$I = \frac{bh^3}{12} = \frac{(1.50)(.50)^3}{12}$$

$$I = .01563 \text{ in}^4$$

From Table 6.05A, Case A (but neglecting beam weight W_B):

$$f_n = 324.96\sqrt{\frac{EI}{W_m L^3}} = 324.96\sqrt{\frac{(29{,}600{,}000)(.01563)}{(10)(12)^3}}$$

f_n = 1681 CPM = 28.0 Hz (neglecting beam weight)

Part B - What is natural frequency if the Beam Weight is included?

Beam Wt = Density X Volume = (.283 lb/in^3) [(12)(1.50)(.50)in^3]

Beam Weight = 2.547 lb (12" Long, 1.5" Wide, .5" Deep)

Now including beam weight (W_B) and using formula from Table 6.05A, Case A:

$$f_{nB} = 324.96\sqrt{\frac{EI}{L^3(W_m + 0.24W_B)}} = 324.96\sqrt{\frac{(29{,}600{,}000)(.01563)}{(12)^3[10 + (0.24)(2.547)]}}$$

f_n = 1632 CPM = 27.2 Hz (3.0% lower when including Beam Weight)

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Part B of example 1 shows that including the beam weight did drop the calculated natural frequency 3.0% from 1681 to 1632 CPM. Therefore, in this case, including the beam weight made only a slight difference.

Part C - What is the beam deflection (D) if you neglect the beam weight?

From Equation 6.05B,

$$f_n = \frac{60}{2\pi}\sqrt{\frac{g_c}{\Delta}}$$

Solving Equation 6.05B for Deflection (Δ),

$$\Delta = \frac{3600 g_c}{(2\pi f_n)^2} \qquad \textbf{(Eqn. 6.05C)}$$

Substituting:

$$\Delta = \frac{(3600)(386)}{(2\pi \text{ X } 1681)^2}$$

$\Delta = .0125 \text{ in} = 12.5 \text{ mils}$ (neglecting beam weight)

NOTE: If you include beam weight, D = .0132 in = 13.2 mils (including beam weight)

Checking Answer C with Cantilever Beam Deflection Formula (neglecting beam weight):

$$\Delta = \frac{PL^3}{3EI} \qquad \textbf{(Eqn. 6.05D)}$$

$$\Delta = \frac{(10)\ (12)^3}{(3)(29{,}600{,}000)(.0156)} = .0125 \text{ in} \quad \text{(OK)}$$

Similarly, natural frequencies can be approximated for machines with shafts mounted between bearings or those with overhung rotors using the formulas provided in Table 6.05A.

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6.06 MECHANICAL LOOSENESS

Many texts and seminars today simply refer to a general term called “Mechanical Looseness” and that it either can be detected by high vibration at 2X RPM or multiple running speed harmonics. However, a comprehensive study of a great amount of available literature as well as review of a number of case histories accomplished by Technical Associates actually has revealed that there are at least 3 different types of Mechanical Looseness, each of which has its own characteristic vibration spectra as well as vibration phase behavior. A discussion will follow on each of these which are as follows:

6.061 *Type A* - Structural Frame/Base Looseness (Primarily 1X RPM);

6.062 *Type B* - Looseness Due To Rocking Motion Or Cracked Structure/Bearing Pedestal (Primarily 2X RPM);

6.063 *Type C* - Loose Bearing In Housing or Improper Fit Between Component Parts (Multiple Harmonics Due To Nonlinearity Often Induced By Impulse Events)

One of the important facts about each type of mechanical looseness is that it alone is not a cause of vibration. Instead, looseness is a reaction to other problems which are present such as unbalance, misalignment, eccentricity, bearing problems, etc. Resolution of these other problems often will remove many of the symptoms, and therefore the response of looseness. However, the problem is that only minute amounts of such problems as misalignment or unbalance can cause vibration if a looseness condition exists. Looseness aggravates the situation. Therefore, mechanical looseness allows much more vibration than would otherwise occur from these other problems alone. Resolution of the other problems themselves will often remove most of the symptoms due to looseness. However, this is often virtually impossible in reality because such steps would require extraordinary levels of precision of alignment or balancing. Therefore, in these cases, the looseness condition will first have to be resolved. Then, if remaining vibration is still high, other steps such as alignment and balancing can be accomplished with much greater ease than before the looseness conditions were resolved. Following below is a discussion on each of the 3 types of looseness which were listed previously:

6.061 Type A - Structural Frame/Base Looseness (1X RPM):

This type of looseness includes each of the following problems:

- Structural Looseness/Weakness of Machine Feet, Baseplate & Concrete Base;
- Deteriorated or Crumbled Grouting;
- Distortion of Frame or Base (Soft Foot);
- Loose Hold-Down Bolts.

Type A Looseness Problems are often misdiagnosed as unbalance or misalignment problems since they have almost identical vibration spectra. Therefore, it is important to look beyond vibration spectra and compare relative amplitudes between directions, look closely at phase behavior which departs radically from such things as unbalance, and to examine other characteristics which are listed as follows:

1. Type A Looseness spectra are dominated by high 1X RPM vibration and appear identical to that for an unbalance or eccentric rotor condition. A spectrum illustrating this type signature is shown in Figure 6.06A.

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2. Often, high vibration is pretty well confined to only one rotor (that is, the driver or driven component or the gearbox alone). This is unlike unbalance or misalignment in which rather high vibration levels due to these other problems are not confined to just one of the rotors.

3. Two different phase behaviors can occur with Type A Looseness:

   a. When comparing vertical and horizontal phase on each of the bearing housings, the vibration will sometimes be found to be highly directional with phase differences of either 0° or 180° depending on whether or not the horizontal reading was taken on one side or the other (either a phase difference of 0° or 180° means that the motion is directly up and down or side to side). This does not normally occur with simple unbalance in which horizontal and vertical phase usually differs approximately 90° (±30°).

   b. When this first phase behavior occurs (0° or 180° phase difference in horizontal and vertical), the analyst should not confine his measurements to the bearing housings alone, but move on down to the machine foot, baseplate, concrete base and surrounding floor. This is illustrated in Figure 6.06B. Here, comparative amplitude and phase measurements should show relatively identical amplitude and phase at 1X RPM at each location. If there is a great difference in amplitude and phase, this will suggest relative motion. Using the point where this great phase change occurs, one can locate where the problem exists. For example, the measurements in Figure 6.06B show a problem between the baseplate and concrete base indicated by the great difference in phase (note 180° out of phase with the other two measurements). This indicates structural looseness/weakness allowing relative movement in machine components which may be due to a problem with the grouting between the baseplate and concrete base; or broken or cracked foundations, etc. On the other hand, if a great phase difference occurred between machine foot and baseplate, this might suggest looseness of the mounting bolt and/or possible stripped mounting bolt threads. Either of these two problem conditions can cause a great vibration at 1X RPM like that shown in Figure 6.06A on the machine component where this occurs, particularly on the bearing housing directly above this base location.

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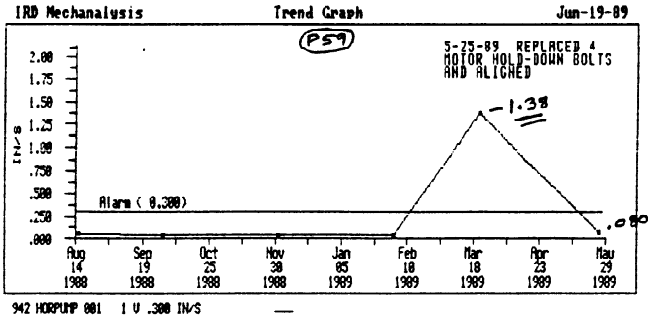
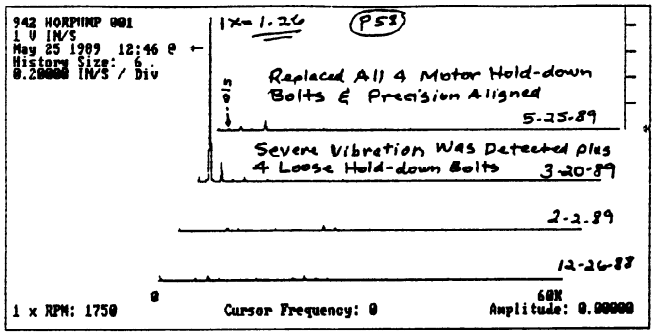


Machine 942 HORPUMP 001 Pos 1 Dir V Units IN/S Alarm .300

Ampl	Insp Code	Date mm/dd/yy	Time	Ampl	Insp Code	Date mm/dd/yy	Time
.062	0	08-14-88	10:01				
.054	0	08-15-88	13:00				
.042	0	09-29-88	11:03				
.038	0	12-01-88	9:49				
.041	0	02-02-89	15:00				
1.38	0	03-20-89	13:43				
.080	0	05-25-89	12:46				



001% x 2.0000 = 0.02000 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
942 HORPUMP 001	1	V	IN/S	May 25 1989 12:46

Frequency	1 x RPM	Amplitude
3600	2.06	0.02478
7200	4.11	0.05782

3% x 2.0000 = 0.06000 1 x RPM: 1800.00

Machine	Pos	Dir	Units	Date
942 HORPUMP 001	1	V	IN/S	Mar 20 1989 13:43

Frequency	1 x RPM	Amplitude
1800	1.00	1.26005
3450	1.92	0.12994

001% x 2.0000 = 0.02000 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
942 HORPUMP 001	1	V	IN/S	Feb 02 1989 13:00

Frequency	1 x RPM	Amplitude
21600	12.34	0.03940

1% x 2.0000 = 0.02000 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
942 HORPUMP 001	1	V	IN/S	Dec 26 1988 10:01

Frequency	1 x RPM	Amplitude
150	0.09	0.02542
21600	12.34	0.02478

FIGURE 6.06A
SEVERE LOOSENESS INTRODUCED BY LOOSE HOLD-DOWN BOLTS

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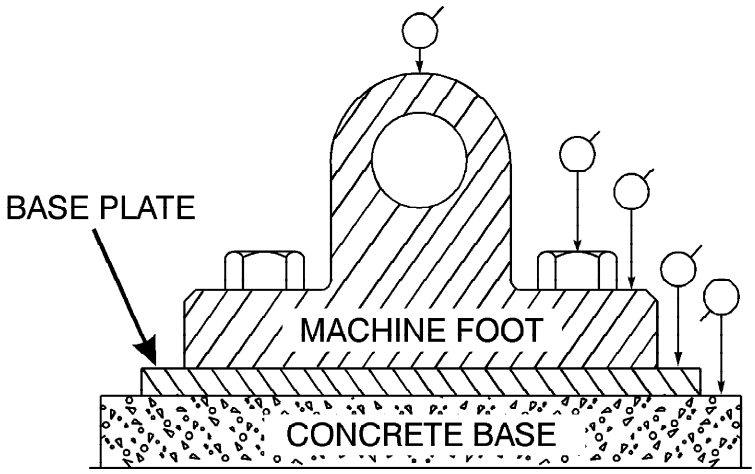

FIGURE 6.06B
PHASE ANALYSIS USED TO PINPOINT LOOSENESS

4. Figure 6.06C illustrates another case involving this high 1X RPM looseness vibration. In this case, bolts that are parallel to the pump shaft are mounted on each pump bearing housing acting in the axial direction. In this case, if these bolts become loose, it will cause a high vibration in the axial direction at 1X RPM which will closely resemble a misalignment problem. However, simply tightening these bolts will greatly reduce the vibration.

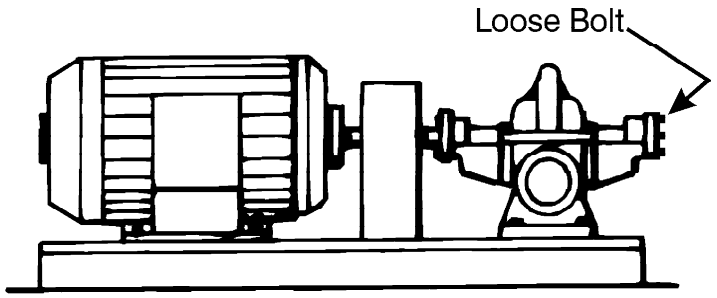

FIGURE 6.06C
EFFECT OF LOOSE BOLT ON PUMP OUTBOARD BEARING HOUSING

5. Distortion induced either by a soft foot or piping strain shows another situation which would result in high 1X RPM vibration signatures looking like unbalance. However, in these cases, when phase readings are taken, they will show highly directional vibration with the difference in horizontal and vertical phase approaching either 0° or 180° (±30°) rather than 90° in the case of simple unbalance. If the problem were distortion rather than looseness, amplitude and phase measurements would show the machine foot, baseplate and concrete base pretty much vibrating in the same direction (equal phase readings). However, they may show that the amplitude on either one of the foot bolts alone is much higher than that on any of the other 3 bolts; or, for example, the right front foot and left rear foot may be significantly higher than the left front and right rear feet. In this second case, the motor would be vibrating diagonally. Great amplitude differences on these foot bolts would suggest a soft foot which must be corrected to reduce the high vibration levels. (In fact, the analyst may find that if he backs off slightly on those bolts having high vibration, he may see much lower vibration than before).

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6.062 Looseness Due To Rocking Motion or Cracked Structure/Bearing Pedestal (2X RPM):

The 2X RPM "looseness symptom" referenced in many vibration texts seems only to occur for the following looseness problems:

- Crack in Structure or Bearing Pedestal;
- Rocking Motion Sometimes Induced By Differential Length Support Legs;
- Occasionally on Some Loose Bearing Housing Bolts;
- When Loose Bearing or Improper Component Fit Problems Are Only of Minor Severity (No Impulse or Impact Events).

Figure 6.06D shows a spectrum typically showing these problems. These problems exhibit the following characteristics:

1. Typically, these problems are suggested when the amplitude at 2X RPM exceeds about 50% of that at 1X RPM in the radial direction.

2. Amplitudes are somewhat erratic.

3. If phase readings are taken with a strobe light, it will often show 2 reference marks which are slightly erratic.

4. These looseness symptoms will not normally occur unless there is some other exciting force such as unbalance or misalignment. However, if this looseness condition exists, it will be extremely difficult to balance or align the unit sufficiently to bring down the final vibration sufficiently.

5. If the looseness problem is a bearing loose in the housing or a loose component on the shaft, the vibration will pretty well remain at 1X and 2X RPM until it worsens allowing an impulse or impact event. When this occurs, these impulses cause nonlinearities in the time waveform which will begin exciting many harmonics advancing to Type C Looseness.

6.063 Loose Bearing In Housing or Improper Fit Between Component Parts (Multiple Harmonics Due To Nonlinearity Often Induced By Impulse Events):

Each of the following problems occur in Type C Looseness:

- Bearing Loose in Housing;
- Excessive Internal Bearing Clearances;
- Bearing Liner Loose In Its Cap;
- Loose Rotor;
- Bearing Loose and Turning on Shaft.

Figures 6.06E thru 6.06G illustrate typical spectra indicating Type C Looseness. Note the presence of multiple running speed harmonics in both spectra. Also note in Figure 6.06G that one of the running speed harmonics lies close to a natural frequency causing a resonant response at this frequency (if the looseness condition is resolved, the resonance condition will likely be as well). **Type C is the most common mechanical looseness problem** and exhibits the following characteristics:

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CASE VI

This case is on our K-1 refrigeration compressor located at our Alkylation Plant. The turbine and compressor both run between 2,700-3,100 RPM requiring 6,000 HP. In this signature (Figure 13) there is a high two times running speed frequency showing up on the turbine inboard. The vibration in this case was suspected to be caused by a looseness problem between the bearing and the bearing housing. The bearing housing nuts were found to be slightly loose and they were tightened. The signature also shows the level had dropped significantly after the housing was tightened; a slight misalignment problem might still be indicated.

Vibration signatures of Turbine IB bearing showing highest level at two times running speed. Bottom signature is before bearing cap is tightened down and top is after.

FIGURE 6.06D
EXAMPLE OF LOOSENESS OF FASTENERS OCCURRING AT 2X RPM
(Ref. 17)

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1. Multiple running speed harmonics sometimes up to 10X or even 20X RPM are clearly present in this spectrum. These harmonics are a result of impulses and truncation (limiting) in the machine response. This impulse event causes a nonlinearity in the time waveform. When this occurs, multiple harmonics will appear in the resulting FFT spectrum (see Figure 6.06E).

2. This looseness tends to produce vibration that is directional which differs from unbalance. It normally will be highest in the direction and vicinity of the looseness problem. For example, this may show that the highest vibration is not either horizontal or vertical, but somewhere in between the two.

3. If the amplitude of harmonics becomes significant, this can also generate frequencies spaced at 1/2 times RPM (that is, .50X, 1.50X, 2.50X, etc.) or even sometimes at 1/3 times RPM.

4. The analyst is cautioned that amplitudes of these 1/2 times RPM "harmonics" may appear deceptively low when compared with those at 1X RPM and running speed harmonics. However, he should remember that no peaks at 1/2 times RPM intervals should be present whatsoever. If the peaks are clearly evident, they do indicate a more advanced looseness problem (or possibly, presence of a rub).

5. One half times RPM "harmonics" usually are accompanied by other problem sources such as unbalance and misalignment.

6. Phase measurements of Type C looseness problems are normally somewhat erratic, but can approach differences of 0° and 180° between horizontal and vertical directions if the vibration itself becomes highly directional. It normally acts in a radial direction, but can occur in the axial, dependent on the exact type of looseness.

7. In the case of a loose rotor such as a loose pump impeller, phase will vary from one startup to the next. The amplitude itself may be steady for a given run, but likewise will vary from startup to startup. Such a loose rotor is impossible to balance since the heavy spot itself is constantly changing directions. This shift in amplitude and phase is likely caused by center of gravity shifts.

8. **CAUTION:** Vibration spectra simply appearing to be Type C looseness (many 1X RPM harmonics) can, in fact, signal a problem of much greater severity - a bearing loose and turning on a shaft. This can be the case even if amplitude of 1X RPM and its harmonics are fairly low, on the order of .05 in/sec or less. In these cases, the turning of the bearing on the shaft can actually cause great damage, actually removing material from the shaft diameter. In these cases, it can cause catastrophic failure of the machine as the bearing finally locks, and can do so without even generating any bearing defect frequencies. Figure 6.06E below is an example of just such a catastrophic failure where over .25 inch of the shaft diameter was removed from a 3.50 inch shaft before the bearing locked up.

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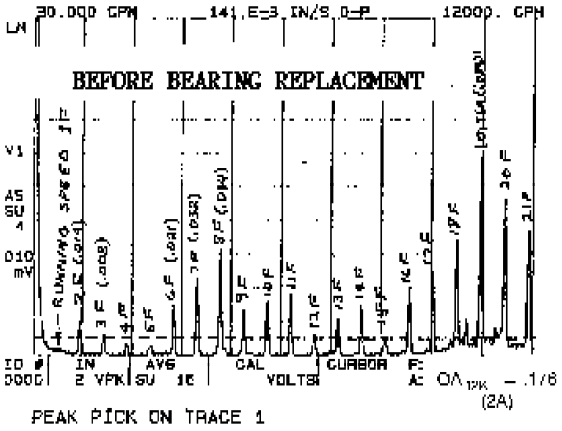


IN/S 0-P	CPM	
13.5E-3	1140.0	-2F
8.37E-3	1710.0	-3F
21.2E-3	3390.0	-6F
32.2E-3	3959.9	-7F
44.4E-3	4530.0	-8F
19.3E-3	5100.0	-9F
22.5E-3	5670.0	-10F
25.8E-3	6240.0	-11F
8.37E-3	6810.0	-12F
15.5E-3	7350.0	-13F
20.6E-3	7920.0	-14F
7.08E-3	8490.0	-15F
28.3E-3	9060.0	-16F
34.8E-3	9630.0	-17F
48.3E-3	10200.	-18F
7.08E-3	10320.	
14.8E-3	10410.	
83.0E-3	10769.	-19F
7.73E-3	10919.	
11.6E-3	11100.	
9.01E-3	11159.	

MILL 4 MAIN DRIVE MOTOR

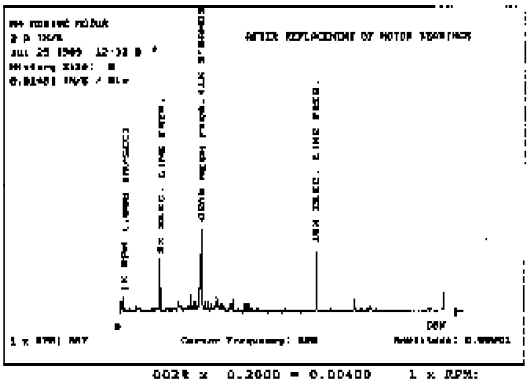
SKF 6326 BEARINGS
BPFI = 6.338 X RPM
BPFO = 4.662 X RPM
BSF = 3.205 X RPM
FTF = 0.424 X RPM

RPM = 567

POINT 2A ON MOTOR
ALL SIGNIFICANT SPIKES
ARE HARMONICS OF
RUNNING SPEED

MECHANICAL LOOSENESS
IS INDICATED

AFTER REPLACEMENT OF MOTOR SHAFT AND BEARINGS
OVERALL = .059 in/sec



NOTE: MOST ALL 1X RPM HARMONICS HAVE BEEN COMPLETELY ELIMINATED

002# x 0.2600 = 0.00400 1 x RPM: 567.00

Machine	Pos	Dir	Units	Date
#4 MDRIVE MOTOR	2	A	IN/S	Jul 25 1999 13:12

Frequency	1 X RPM	Amplitude	Frequency	1 X RPM	Amplitude
150	0.26	0.00550	36000	63.49	0.03050
600	1.06	0.00801	41200	78.13	0.00648
7800	13.70	0.02701	59050	105.56	0.01048
10500	18.52	0.00600			
10950	19.31	0.00400			
12900	22.75	0.00950			
13500	23.81	0.00648			
14700	25.93	0.04149			
15300	26.98	0.00550			
15900	28.04	0.00648			
16500	29.10	0.00448			
17550	30.95	0.01048			
18450	32.54	0.00445			
19950	35.19	0.00448			
20550	36.24	0.00750			
22800	40.21	0.00499			
23400	41.27	0.00499			

T - Threshold adjust Q - Quit

FIGURE 6.06E
BEFORE & AFTER REPLACEMENT OF A DC MOTOR BEARING WHICH WAS TURNING ON THE SHAFT RESULTING IN CATASTROPHIC FAILURE

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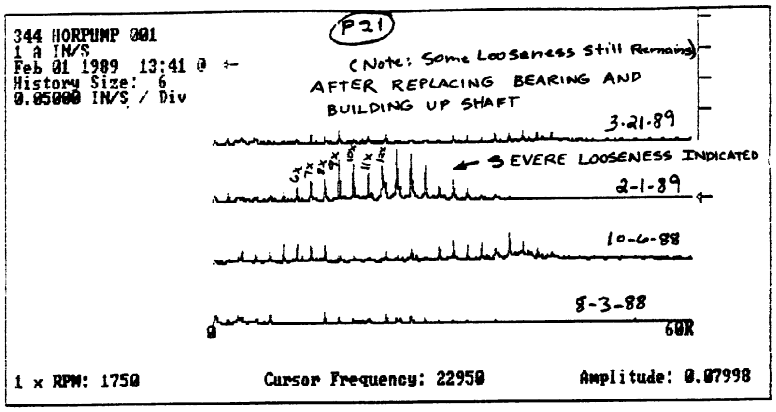

2% × 0.5000 = 0.01000 1 × RPM: 1750.00

Machine	Pos	Dir	Units	Date
344 HORPUMP 001	1	A	IN/S	Mar 21 1989 12:57

Frequency	1 × RPM	Amplitude
12300	7.03	0.01247
15900	9.09	0.01843
19500	11.14	0.01048
21600	12.34	0.01096
35250	20.14	0.01144
37050	21.17	0.01191
38850	22.20	0.01493
40650	23.23	0.01295
42450	24.26	0.01096

006% × 0.5000 = 0.03000 1 × RPM: 1750.00

Machine	Pos	Dir	Units	Date
344 HORPUMP 001	1	A	IN/S	Feb 01 1989 13:41

Frequency	1 × RPM	Amplitude
12300	7.03	0.03002
14100	8.06	0.03503
15900	9.09	0.06497
17700	10.11	0.05997
19500	11.14	0.04249
21150	12.09	0.05496
22950	13.11	0.07998
24750	14.14	0.07498
26550	15.17	0.05496
30000	17.14	0.03002

2-1-89

002% × 0.5000 = 0.01000 1 × RPM: 1750.00

Machine	Pos	Dir	Units	Date
344 HORPUMP 001	1	A	IN/S	Aug 03 1988 07:27

Frequency	1 × RPM	Amplitude
300	0.17	0.01398
3000	1.71	0.01096
7200	4.11	0.01096
14100	8.06	0.01597
21600	12.34	0.01247

FIGURE 6.06F
ADVANCING PROBLEM WITH BEARING LOOSE ON SHAFT
(PARTIALLY RESOLVED)

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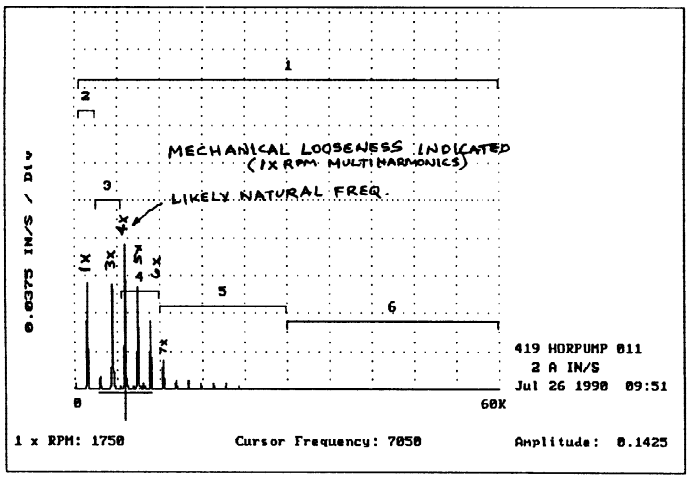

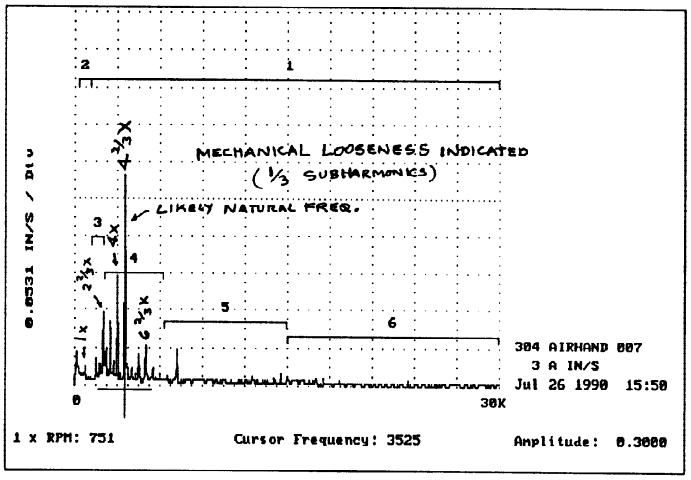

002% x 0.500 = 0.0100 1 x RPM: 751.00

Machine	Pos	Dir	Units	Date
304 AIRHAND 007	3	A	IN/S	Jul 26 1990 15:50

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
150	0.20	0.0450	5550	7.39	0.0225 = 7⅓X
750	1.00	0.0275 = 1X	5700	7.59	0.0100
975	1.30	0.0100 = 1⅓X	6000	7.99	0.0150 = 8X
1125	1.50	0.0100 = 1½X	6525	8.69	0.0125 = 8⅔X
1500	2.00	0.0350 = 2X	7200	9.59	0.0500 = 9⅔X
1800	2.40	0.0300	12525	16.68	0.0125 = 16⅔X
2025	2.70	0.1025 = 2⅔X	14025	18.68	0.0100 = 18⅔X
2250	3.00	0.0525 = 3X	14550	19.37	0.0150 = 19⅓X
2475	3.30	0.0900 = 3⅓X	15000	19.97	0.0100 = 20X
2775	3.70	0.0325 = 3⅔X	17025	22.67	0.0100
3000	3.99	0.1550 = 4X	17550	23.37	0.0100
3225	4.29	0.0150 = 4⅓X			
3525	4.69	0.3000 = 4⅔X			
4050	5.39	0.0225 = 5⅓X			
4275	5.69	0.0150 = 5⅔X			
4500	5.99	0.0425 = 6X			
5025	6.69	0.0575 = 6½X			

T - Threshold adjust Q - Quit

FIGURE 6.06G
2 DIFFERENT TYPES OF SPECTRA INDICATING
TYPE C MECHANICAL LOOSENESS

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6.07 ROTOR RUB

Rotor rub occurs when the rotating parts of a machine contact stationary parts and can be catastrophic, depending on specifically which parts contact one another, the material construction of each, rotational speed-induced friction, structural rigidity of both the rotating and stationary parts and the impact velocity itself. Technical papers, including reference 6, have classified two major types - (1) Partial Rub, and (2) Full Annular Rub. Each of these cases will be separately discussed in Sections 6.071 and 6.072, respectively. Rotor rub encompasses a variety of events, some of which are catastrophic, and others which are less catastrophic (but still potentially serious). These include the following examples:

CATASTROPHIC RUB EVENTS

Journal Contacting Bearing Babbitt

Rotor Contacting Stator in Electric Motor

Impeller Vanes Contacting Diffuser Vanes

Turbine Blades Contacting Diffuser Vanes

LESS CATASTROPHIC RUB EVENTS

Shaft Rubbing a Seal

Coupling Guard Rubbing a Shaft

Belt Rubbing Belt Guard

Fan Blades Contacting Shroud

Characteristics which describe rotor rub in general are as follows:

1. When rub occurs, it can actually change the stiffness of the rotor which will therefore change the rotor natural frequency during the rub event. This can become a real problem if the rotor was formerly operating in a nonresonant condition, but now is thrust into resonant operation due to the natural frequency change.

2. The impact can cause local wear and deformation depending on the angle of attack and impact velocity.

3. Not only can high forces be generated due to impact, but also high torques which can almost instantaneously affect electrical characteristics, sometimes causing a significant increase in amperage. Of course, if the rub is continuous, this can bring about failure, not only of the contact area, but also potentially of the driver itself.

4. When rub occurs, friction will oppose rotation depending on the amount of force and surface properties such as coefficient of friction. This high friction force can even cause a backwards whirling motion, sometimes known as “backward precession”. This does not mean the shaft changes rotational direction, but backward whirling implies that the shaft can actually begin to orbit in a direction opposite shaft rotation at a frequency independent of rotational speed.

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5. A Partial Rub usually precedes a Full Annular Rub.

6. Waterfall plots (also called Cascades) are especially helpful in diagnosing radial rubs. Example Cascade plots are shown in Figure 6.07A which illustrates a partial rub. These cascade plots show how vibration amplitude and frequency changes with machine RPM. Looking at Figure 6.07A, this figure shows a series of single spectra plotting amplitude versus frequency, and it also shows how they change as the machine is brought up to speed (note that the 1X RPM goes through a natural frequency at about 1500 to 1600 RPM. Later, the amplitude at 1/2 times RPM goes through the same natural frequency, at which point the amplitude 1/2 times RPM is much greater than that at 1X RPM when the speed of the machine is approximately 3600 to 3800 RPM. This is very unusual and, when 1/2 times RPM is much greater than that at 1X, rub is one of the primary suspects.

7. Shaft orbit displays, which are also shown in Figure 6.07A, are also very helpful in diagnosing rubs. These orbit displays show the actual path the shaft itself follows inside the bearing. The shape of the orbit is helpful in determining phase as well as distinguishing the nature of the rub, its severity, and probable frequency content.

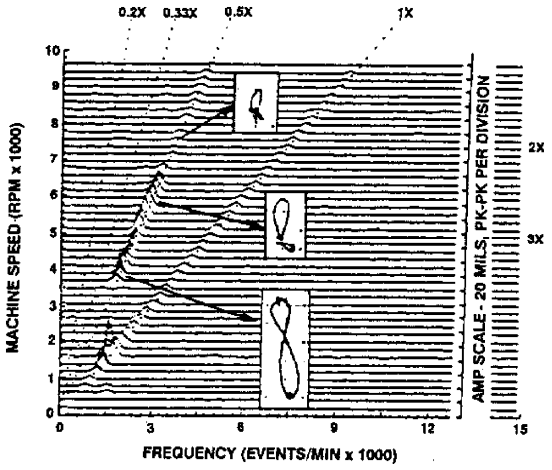

FIGURE 6.07A
CASCADE DIAGRAM INDICATING A PARTIAL RUB EVENT UNDER HIGH FORCE (Ref. 6, pg. 3)

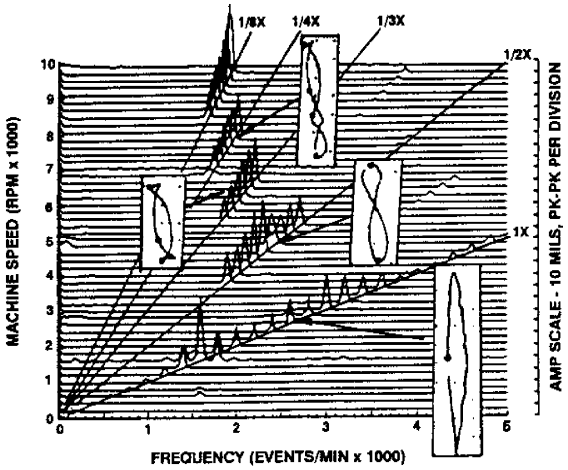

FIGURE 6.07B
PARTIAL RUB EVENT WITH LOW RUB FORCE GENERATING MANY INTEGER SUBHARMONICS (Ref. 6, pg. 3)

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8. Again, two distinct types of radial rubs will occur including partial rub and full annular rub:

    a. Both types can take place separately or simultaneously;

    b. Both types originate either from excessive shaft vibration, from a smaller vibration if the shaft is not rotating in its geometric centerline, or if insufficient clearances have been established between rotating and stationary parts.

While many things are shared in common between partial and full annular rubs, each type has its own unique characteristics which are summarized in Sections 6.071 and 6.072, respectively.

6.071 Partial Rub:

Partial Rubs usually begin only as very short-lived, transient events which often grow in severity and in duration as time passes. Because most of these rub events are only transient, the backwards whirling or precession motion does not normally occur. Left uncorrected, localized heating will occur which can actually bow the shaft and cause even greater severity rubs and cause rotor instability. Of course, as more severe rubs occur, even greater heat is generated in the shaft bowing it further which can lead to eventual catastrophic failure. Therefore, it is important to recognize partial rubs and act on them before they are given the chance to become even more serious. Following below are some of the characteristics exhibited by a partial rub which will help the analyst recognize a partial rub event:

1. Partial rubs will generate noticeable subsynchronous vibration (vibration below 1X RPM) which can sometimes be of significant amplitude and often will excite high frequencies as well if any noticeable impact occurs. However, an important fact is that subsynchronous vibration itself should not occur. Just the fact that subsynchronous vibration is occurring is important, and its source needs to be identified. If noticeable impact does occur, it can excite the system natural frequencies (independent of speed) in the mid to high frequency ranges.

2. If the subsynchronous vibration is caused by partial rub, it will normally show integer sub harmonics (1/2X or 1/3X or 1/4X, etc.). Which integer sub-harmonic is excited depends on the shaft speed relative to rotor system natural frequency. In general, with higher shaft speed, lower fractions of subharmonic vibration can occur.

3. Several different integer subharmonics can simultaneously occur when a light rub with lower radial force occurs. Such a response is shown in Figure 6.07B which shows both the spectra and the orbit responses for 1X, 1/2X, 1/3X, 1/4X, and 1/5X RPM.

4. Often, when a heavier radial force partial rub occurs, only one subharmonic is present - predominantly, 1/2X RPM will remain.

5. If sharp impacts occur, multiples of these subharmonic frequencies can also appear (.5X, 1X, 1.5X, 2X, 2.5X, etc.) as well as high frequency resonant response. This is due to inherent nonlinearities which occur in an FFT spectrum when sharp impact takes place (instead of the time waveform being a smooth sinusoidal shape, it is truncated during an impact event which can generate almost a square waveform, resulting in a series of running speed peaks on an FFT spectrum). Such a spectrum is seen in Figure 6.07C with a whole series of 1/2 X RPM subharmonics which occurred when blading of a reactor began rubbing the stainless steel tank wall. Note that Figure 6.07D then compares “before” and “after” spectra taken prior to and after the repair actions at 9 RPM speed, respectively.

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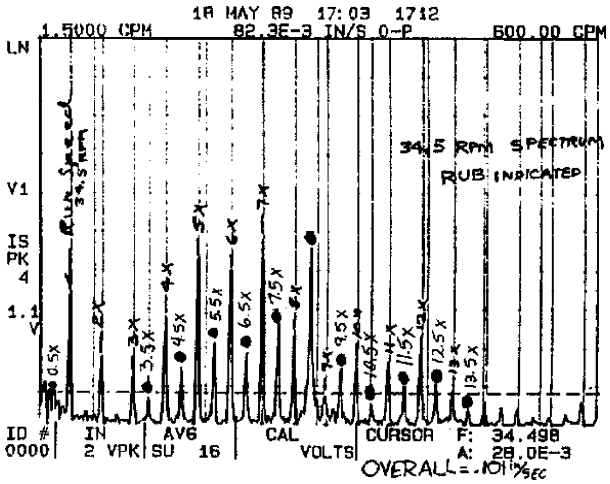

FIGURE 6.07C
SPECTRUM CAPTURED DURING THE CONFIRMED RUB OF A CHEMICAL REACTOR BLADE AGAINST ITS STAINLESS STEEL WALL WITH BLADE SPEED OF 34.5 RPM (Notice 1/2X RPM Harmonics Identified By Dots)

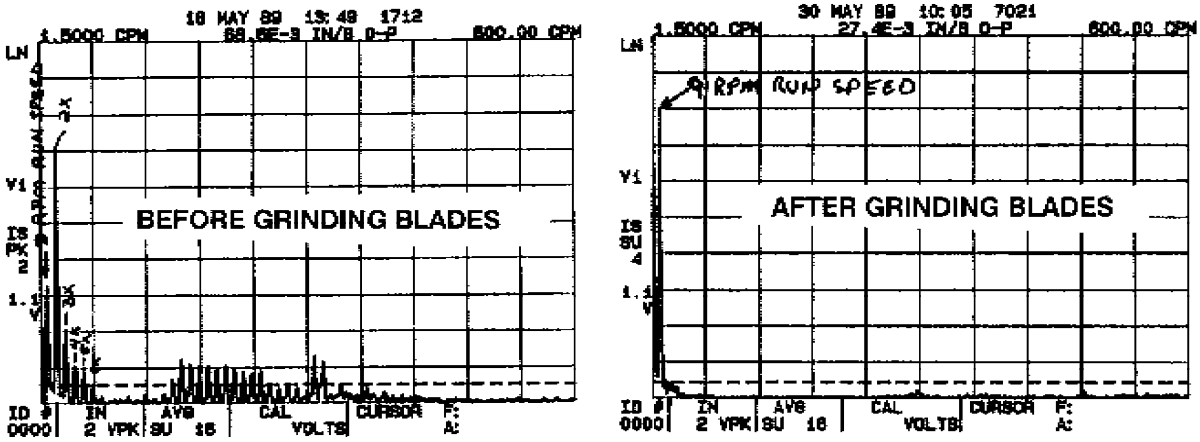

FIGURE 6.07D
COMPARISON OF SPECTRA BEFORE AND AFTER REPAIR RESOLVING RUB PROBLEM (BOTH SPECTRA CAPTURED AT 9 RPM REACTOR SPEED)

6. During a partial rub, the response is highly directional which results in horizontal and vertical phase difference of subsynchronous vibrations approaching 180° between horizontal and vertical. This is seen in Figure 6.07B. Note the elongated shape of the orbit at 1/2X RPM as well as that at 1/4X RPM, each of which approach straight line motion with 180° phase difference between the horizontal and vertical proximity probes.

6.072 Full Annular Rub:

As previously mentioned, partial rub events can eventually lengthen in duration and impact leading to full annular rub in which case virtually a continuous rub event occurs. A full rub may cause counter-rotation, also known as backwards precession during which the shaft will continue to rotate in the same direction, but will orbit in a direction opposite this. This backward precession is caused by friction applied to the rotor generating a force in the opposite direction

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at the point of contact. This force actually creates a torque in the opposite direction which can become very large not only detrimentally affecting the health of the machine, but also affecting energy requirements. In general, full annular rubs are characterized by each of the following characteristics:

1. Figure 6.07E shows a spectrum taken during a full annular rub along with its shaft orbit displays when a seal was rubbing on a shaft. This shows how the response changed as the machine was brought up to speed. Note that at lower speeds little vibration was noticed until the rotor was up to approximately 2500 to 3500 RPM. At these speeds, the rotor began bouncing around the seal as seen by the orbit display. At higher speeds on the order of 3500 RPM, a response at about 7000 CPM began which remained at this same frequency even when the machine was brought all the way up to 5000 RPM. This 7000 CPM peak corresponded to the rotor/seal system natural frequency which was higher than the natural frequency of the rotor alone before it contacted the seal due to the additional stiffness supplied by the seal during the rub event. This 7000 CPM frequency then was the "speed" of the backward precession. That is, this backward precession will easily initiate at resonance and will remain at the same frequency location, independent of shaft rotational speed, whether or not the speed is increased or decreased.

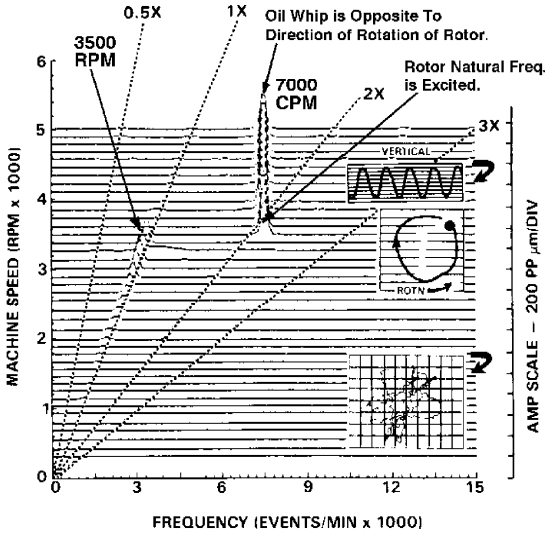

FIGURE 6.07E
MULTIPLE IMPACTING SEAL RUB WHICH GENERATED REVERSE PRECESSION AT APPROX. 3500 RPM EXCITING A 7000 CPM ROTOR/SEAL SYSTEM NATURAL FREQUENCY (Ref. 17, Pg. 19)

2. Backward precession is inherently unstable and can be very violent causing catastrophic rotor destruction.

3. Therefore, a full annular rub is characterized by a forward precession at 1X RPM corresponding to the unbalance in the system which can lead to backward precession at the natural frequency of the rotor/contact surface system (in which case both 1X RPM and the rotor/contact surface natural frequency will appear, often with 1X RPM at much lower amplitude).

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4. The catalyst which causes self-excited backward precession is friction between the rotating and stationary components. Since friction itself is nonlinear, it also can generate a series of running speed harmonics. Here again, higher harmonics are generated in an FFT spectrum by time waveform nonlinearities.

5. Figure 6.07F shows another type of full annular rub which is dominated by the response at 1X RPM along with some running speed harmonics at somewhat lower amplitudes.

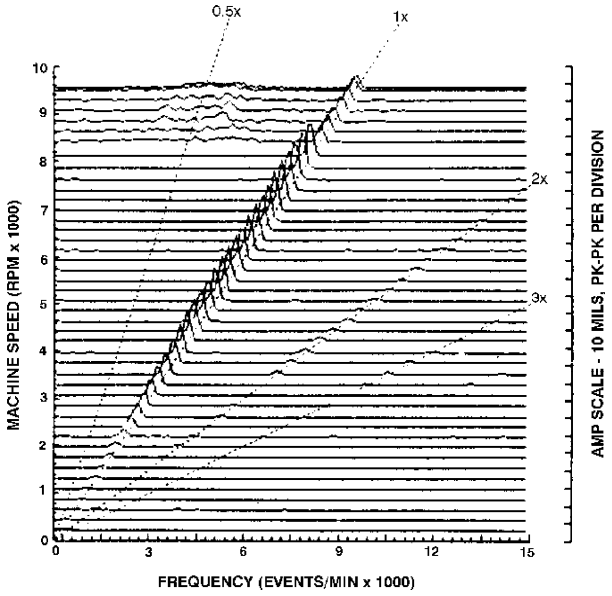

FIGURE 6.07F
FULL ANNULAR RUB DOMINATED BY FORWARD PRECESSION SYNCHRONOUS SPEED (Ref. 6, Pg. 4)

6. Figures 6.07G and 6.07H are examples of a full annular rub where self-excited backward precession has occurred at a speed just below the first balance resonance. Note that the backward precession remained throughout the runup and even continued to occur as the machine coasted back down in Figure 6.07H. Note that when the backward precession was occurring, the comparative amplitude at 1X RPM was very low.

7. Note that since friction remains continually throughout a full annular rub, it will often introduce nonlinearities generating many running speed harmonics as well.

8. The vibration amplitude which will result in a full annular rub depends on clearances and damping of the system.

Permanent monitors with horizontal and vertical probes set 90° apart are recommended on all critical machinery in which rub can occur. This normally is thought of to be very large turbo-machinery and compressors outfitted with sleeve bearings. However, rotor rub events are not confined to these machines. They also can occur on smaller machinery and on units outfitted with antifriction bearings. Therefore, if the machine is truly critical, and there is a real possibility of a rub event occurring, these machines should also be outfitted with permanent vibration monitoring. This will allow not only capture of spectra during normal operation, but will allow the analyst to capture both waterfall spectra and shaft orbit displays similar to that shown in Figure 6.07A during startups and coastdowns. Not only will this go a long way in improving chances for detecting rubs when they do occur, but also permanent monitors will provide protection against undetected, catastrophic failure when no measurements are being made, particularly in those situations where the whole rub and catastrophic failure event can occur within minutes, or even seconds.

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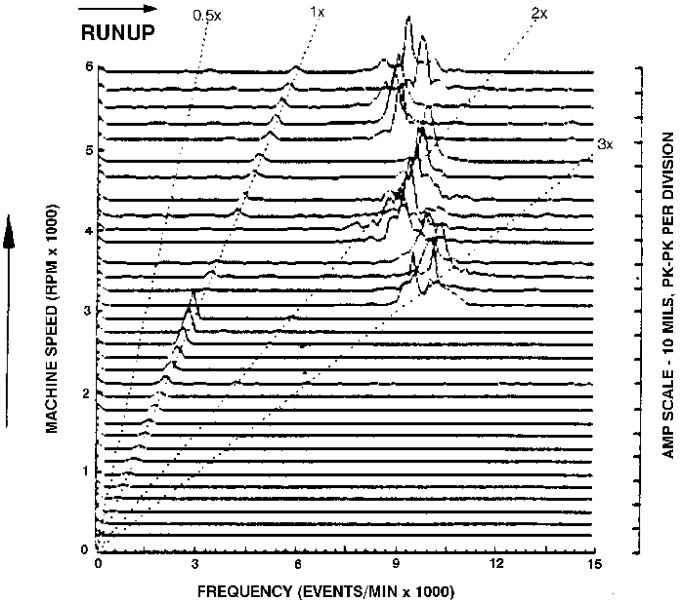

FIGURE 6.07G
FULL ANNULAR RUB DURING RUNUP WITH SELF-EXCITED BACKWARD PRECESSION (Ref. 6, Pg. 5)

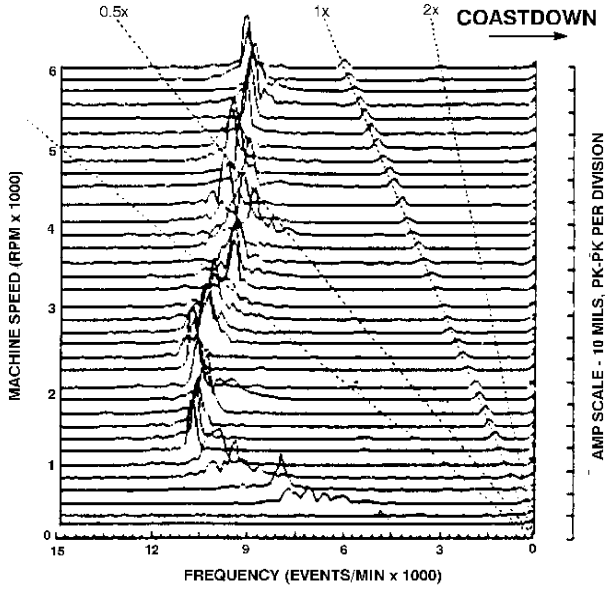

FIGURE 6.07H
FULL ANNULAR RUB DURING COASTDOWN (NOTE CONTINUED BACKWARDS PRECESSION UNTIL VERY LOW SPEED) (Ref. 6, Pg. 5)

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6.08 JOURNAL BEARINGS

Journal Bearings are also sometimes called Hydrodynamic Journal Bearings (technically probably a better name), sleeve bearings, Fluid Film Bearings or Plain Bearings. There are a variety of these bearings, some of which are listed in Figure 6.08A. As per Reference 10, "Hydrodynamic Bearings operate by developing a fluid film between the rotating shaft journal and the stationary bearing bore". Figure 6.08B shows the hydrodynamic bearing pressure profile and how it is distributed relative to the rotating shaft. Note in this figure that the shaft is rotating counterclockwise and that the maximum pressure is located roughly 15° to 20° counterclockwise of vertical. This is common in fluid film bearings.

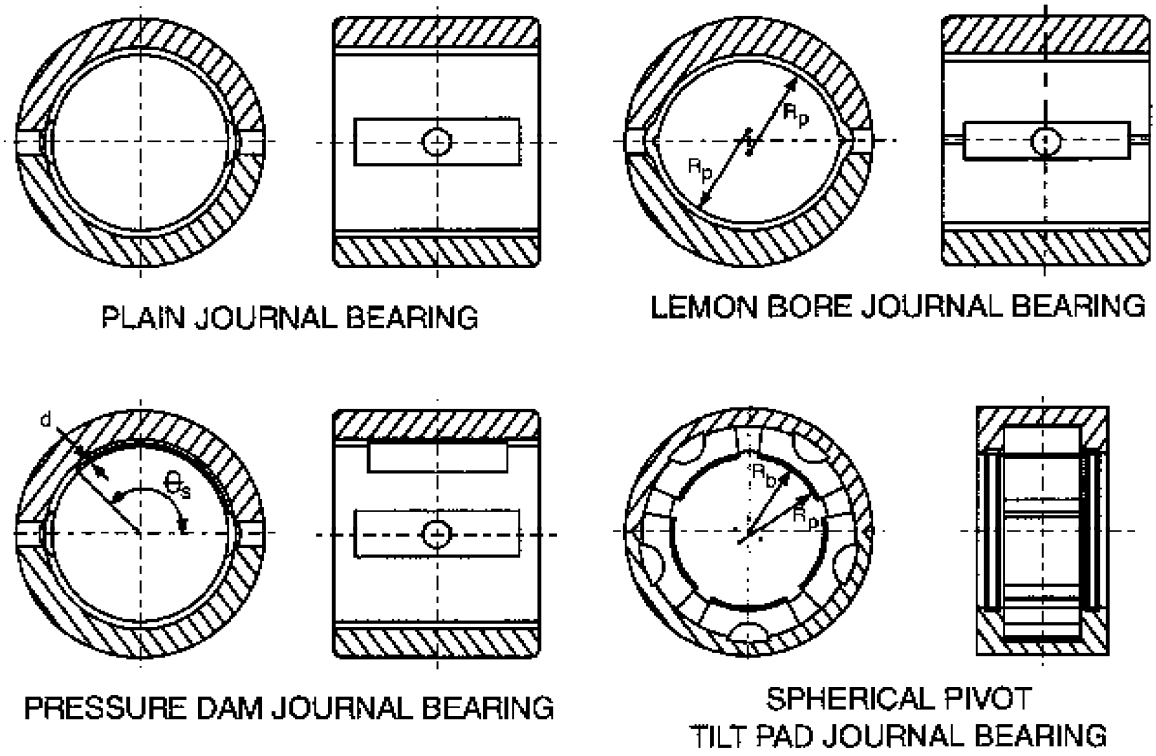

FIGURE 6.08A
VARIOUS TYPES OF HYDRODYNAMIC JOURNAL BEARINGS
(Reference 10, Pages 48, 49, 50 & 53)

Figure 6.08C helps demonstrate one of the key items in a journal bearing - that of stability. The offset between the centerline of the bearing and that of the journal (shaft) is known as the bearing eccentricity. Importantly, eccentricity decreases as load decreases as oil viscosity (μ) increases and/or as speed increases. Note the dashed line running through each of the centerlines. Note on the drawing that this is referred to as the "line of centers" which is used to determine the "attitude angle" which is the angle between the line of centers and the applied load vector. These two quantities, eccentricity and attitude angle, are key indicators of bearing stability. Note that as the bearing load drops and/or the rotor speed increases, the attitude angle increases in plain cylindrical journal bearings. As this attitude angle increases, the bearing stability will decrease. Rotor instability occurs when a bearing is unable to exert sufficient preload to keep the rotating shaft in a stable position or as the rotor-bearing system loses its damping capabilities. When a rotor bearing system is prone to becoming unstable, any outside force which acts to upset the

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bearing load may provide the condition necessary to cause instability (in some cases, a rotor-bearing system may suddenly become unstable even if there is no outside disturbance). Instability is a condition of rotor operation in which various elements in the system combine to induce self-excited vibration that can remain even after the original stimulus has been removed (self-excited vibrations are due to an internal feedback mechanism transferring the rotational energy into vibration). Normally, only a significant reduction in operating speed will again stabilize the rotor. In these cases, the rotor will often stabilize allowing it to be returned to its normal operating speed, but will often remain stable only until some disturbing force again upsets the system. When the forces restraining instability are reduced or lost, the vibration amplitude may suddenly increase dramatically, but yet stabilize at a value much higher than the amplitude at running speed. At this point, the dominant vibration itself actually is normally occurring subsynchronous to the operating speed (that is, lower than rotating speed). Therefore, one of the key items to watch in vibration signatures on journal bearings will be subsynchronous vibration below running speed.

In general, the greatest concern in the subsynchronous region for machines outfitted with journal bearings is concentrated between 35% and 55% of operating speed. However, particular problems can cause subsynchronous vibration as low as 20% up to as high as 80% of operating speed, depending on the particular problem. Following in Sections 6.081 through 6.084 will be a discussion on some of the more common journal bearing problems which can be detected by vibration analysis including journal bearing wear and clearance problems, oil whirl, instability and oil whip instability.

6.081 Journal Bearing Wear and Clearance Problems:

Worn journal bearings can be detected either by placing a velocity pickup or an accelerometer on the bearing cap. However, probably the best journal bearing condition data at lower frequencies up to approximately 5X RPM is captured from non-contact probes reading relative shaft vibration (also known as proximity probes). These proximity probes are placed just off the shaft surface (approximately 40 to 60 mils depending on shaft diameters and materials and always must be placed within the linear range of the non-contact pickup) and sense shaft motion relative to the probe itself. Therefore, they measure vibration displacement which is probably a good indicator of condition out to about the 4th or 5th running speed harmonic. The key point is that the proximity probe is looking directly at shaft motion, whereas both the accelerometer and velocity pickup on the bearing cap must sense a signal which must pass from the shaft through the oil film, and then through several metal interfaces to the bearing cap surface.

It is important to point out that some journal bearing failure incidents take place in only a matter of minutes, or even seconds, depending on the particular problem and the type of instability to which the system is subjected. However, in those cases where the deterioration does take place over a longer period of time, the oil film characteristics themselves begin to change and eventually may bring about rotor instability. In these cases, worn journal bearings can be detected either directly from the shaft or on the bearing cap.

When taking data from a proximity probe, it is important to point out it is quite normal to see several running speed harmonics. This is unlike velocity spectra taken from bearing caps in which case normally only the first 2 or 3 harmonics are seen, and each succeeding harmonic normally is only about 1/3 the height of the former (if no problems are present). Still, even with proximity probe shaft vibration data, the harmonics should also disappear into the spectral base.

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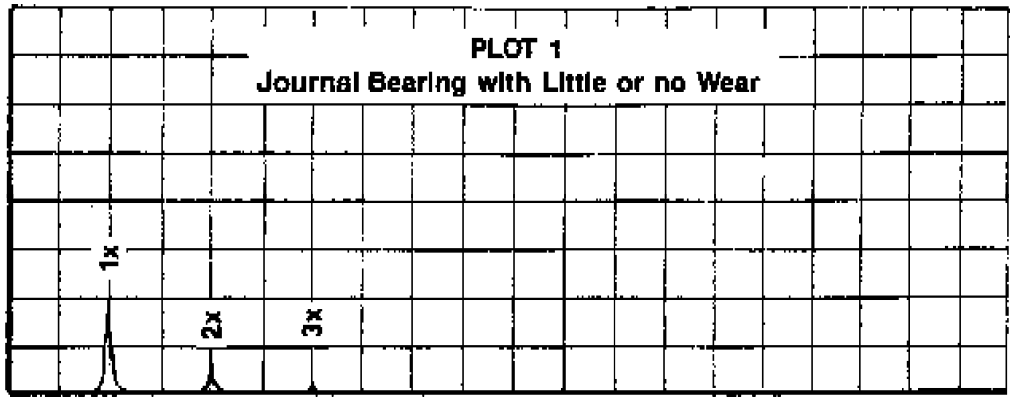
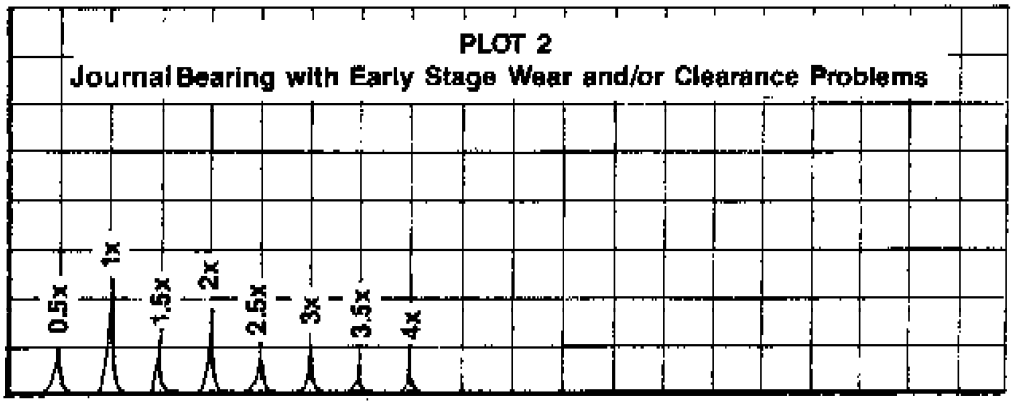
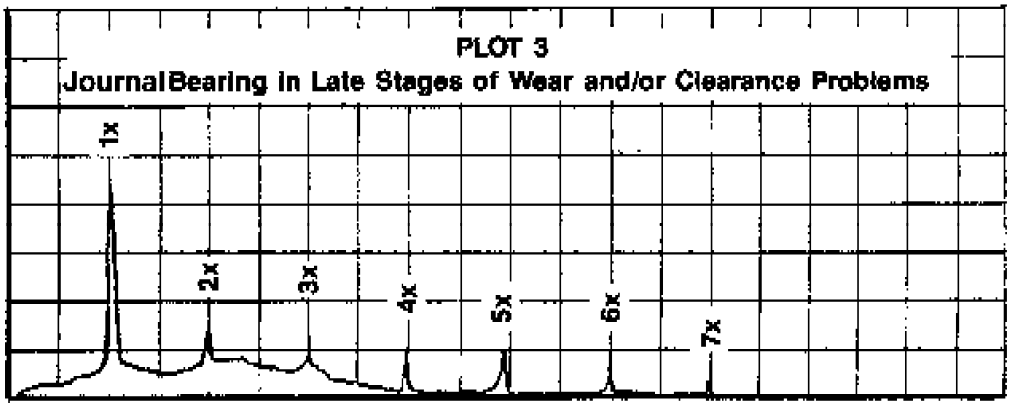
FIGURE 6.08D
TYPICAL SPECTRA SHOWING JOURNAL BEARING PROBLEMS







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Following below are characteristics displayed by journal bearings with clearance and/or wear problems:

1. When high vibration levels are generated by faulty journal bearings, they are generally the result of lubrication problems, improper bearing load, looseness (babbitt loose in the housing), or excessive bearing clearance (caused by wiping or chemical erosion).

2. Figure 6.08D shows typical spectra for a journal bearing with little wear and/or clearance problems, one with early stage problems, and one with significant problems evident. Note the presence of subsynchronous vibration during early stages which may sometimes be at 1/2X or even 1/3X RPM subharmonics. In the latter stages of journal bearing wear, note that the spectrum will typically show a high vibration at 1X RPM, along with harmonics superimposed on an elevated baseline, particularly below 3X RPM.

3. A wiped journal bearing can often be detected by comparing horizontal and vertical vibration amplitudes. Normally, the horizontal vibration should be somewhat higher than that in the vertical direction due to more support in the vertical direction. A wiped journal bearing will often allow abnormally high vibration in the vertical direction as compared to that in the horizontal.

4. A wiped bearing with excessive clearances can allow the shaft to actually change position within the bearing which may result in misalignment which often will generate a vibration at 2X RPM. Sometimes, a bearing with excessive clearance will finally allow the shaft to rub the bearing or possibly another component such as a seal. In these cases, it can be critical in that this may be just the right disturbing force necessary to set the shaft into violent motion causing rotor instability. However, there have been cases reported where misalignment has actually had a stabilizing effect on the journal bearing. This may have been due to the additional load caused by the misalignment acting on a shaft and having a bearing which might have been too lightly loaded to maintain stability.

5. A journal bearing having excessive clearance may allow a small unbalance, misalignment or other related force to result in mechanical looseness, thereby generating the lower vibration spectra of Figure 6.08D. In these cases, the bearing is not the source of the problem. However, the vibration amplitude would be much lower if the bearing clearances were to spec.

6. Some oil film bearings act as thrust bearings. In these cases, many of them have pads or shoes. Of course, when running properly, the shaft should not contact the thrust bearing, but ride on an oil film instead. When problems develop with these thrust bearings, much higher vibration will result. This vibration sometimes will occur at 1X RPM, predominantly in the axial direction. However, if the thrust bearing is outfitted with pads, the vibration will be at the #pads X RPM. Surprisingly, this so-called pad passing frequency (#pads X RPM) can sometimes generate higher radial vibration than that in the axial direction. In any case, when excessive levels do occur at either the pad passing frequency or at 1X RPM, thrust bearing problems should be suspected. Of course, if this occurs at 1X RPM, the question may arise as to whether the problem is unbalance or thrust bearing wear. If this thrust bearing is placed in a compressor, one test that can be performed to detect which problem dominates is simply taking data with the compressor fully loaded and comparing that with the machine unloaded. If the problem is with the thrust bearing, the vibration should drop significantly as the load drops. However, if unbalance dominates, there should be little or no difference in 1X RPM amplitude since you have only dropped the load and not the machine RPM.

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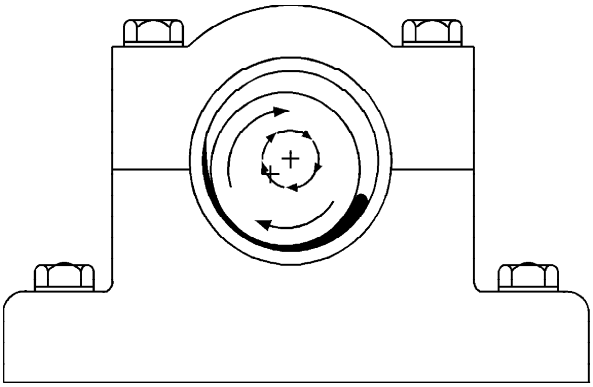

FIGURE 6.08E
OIL FILM WITHIN A JOURNAL BEARING

6.082 Oil Whirl Instability:

Oil Whirl is probably the most common cause of subsynchronous instability in hydrodynamic journal bearings. Normally, the oil film itself will flow around the journal to lubricate and cool the bearing. In so doing, it will develop an average speed slightly less than 50% of the journal surface speed. This is pictured in Figure 6.08E. Normally, the shaft rides on the crest of an oil pressure gradient, rising slightly up the side of the bearing slightly off vertical at a given, stable attitude angle and eccentricity. The amount of rise depends on the rotor speed, rotor weight and oil pressure. With the shaft operating eccentrically relative to the bearing center, it draws the oil into a “wedge” to produce this pressurized load-carrying film.

If the shaft receives a disturbing force such as a sudden surge or external shock, it can momentarily increase the eccentricity from its equilibrium position. When this happens, additional oil is immediately pumped into the space vacated by the shaft. This results in an increased pressure of the load-carrying film which creates additional force between the film and shaft. In this case, the oil film can actually drive the shaft ahead of it in a forward circular motion and into a whirling path around the bearing within the bearing clearance. If there is sufficient damping within the system, the shaft can be returned to its normal position and stability. Otherwise, the shaft will continue in its whirling motion which can become violent depending on several parameters.

Oil whirl will demonstrate the following characteristics:

1. The oil whirl condition can be induced by several conditions including:

    - light dynamic and preload forces;
    - excessive bearing wear or clearance;
    - a change in oil properties (primarily shear viscosity);
    - an increase or decrease in oil pressure or oil temperature; improper bearing design (sometimes an over design for the actual shaft loading);
    - fluid leakage in the shroud of blades and shaft labyrinth seals (so-called "Alford force" or "aerodynamic force");
    - change in internal damping (hysteretic, or material damping; or dry (coulomb) friction);
    - gyroscopic effects - especially on overhung rotors having much overhang.

    Any of the above conditions can induce oil whirl after a disturbing force induces an initial rotor deflection.

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2. Sometimes machines can exhibit oil whirl intermittently that have nothing to do with the condition of the journal bearing, but rather to external vibratory forces transmitting into the unit or from sources within the machinery itself. In these cases, these vibratory forces just happen to have the same frequency as the oil whirl frequency of that bearing and can be just the amount of disturbing force at just the right tuned frequency to set it into the whirl motion. This vibration can transmit from other machinery through attached structures such as piping and braces, or even through the floor and foundation. If this does occur, it may be necessary to either isolate this machine from surrounding machinery, or to isolate the offending machine itself.

3. Oil whirl can be easily recognized by its unusual vibration frequency which is generally on the order of 40% to 48% of shaft RPM (Reference 2 states that pure oil whirl occurs at 43% of shaft speed, but that the instability may occur at the 1st critical speed).

4. Figure 6.08F shows the development of oil whirl just after the shaft is brought up to speed. Note that the shaft definitely went into whirl at a machine speed of about 1800 RPM and remained in whirl until about 4000 RPM. At this point, note that the shaft 1X RPM begin entering resonance which actually caused enough force to overcome whirl. However, once the machine has passed through resonance, whirl once again occurred just above 5200 RPM.

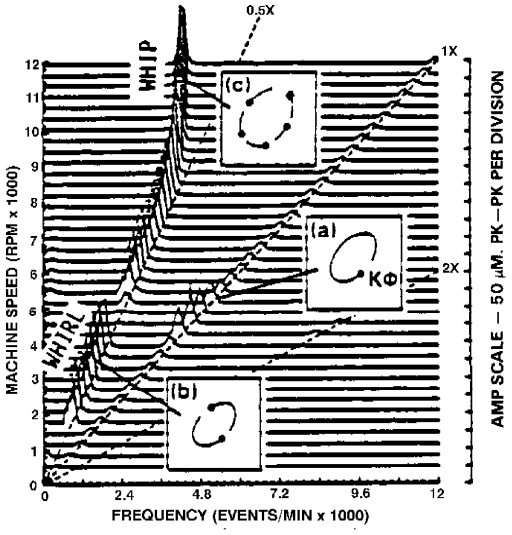

FIGURE 6.08F
DEVELOPMENT OF OIL WHIRL JUST AFTER STARTUP
(FOLLOWED BY OIL WHIP FROM 9200-12,000 RPM)
(Reference 17, Page 20)

5. Oil whirl is considered severe when vibration amplitudes reach 50% of the normal bearing clearance. At this point, corrective action must be taken.

6. Temporary corrective measures include changing the temperature of the oil (and therefore, the oil viscosity), purposely introducing a slight unbalance or misalignment to increase the loading, "temporarily" shifting the alignment by heating or cooling support legs, scraping the sides or grooving the bearing surface to disrupt the lubricant "wedge" or changing the oil pressure.

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7. More permanent corrective steps to solve oil whirl problem include installing a new bearing shell with proper clearances, preloading the bearing by an internal oil pressure dam, or completely changing the bearing type to oil film bearings less susceptible to oil whirl (including axial-groove bearings, lobed bearings or tilting pad bearings). The tilting bad bearing is possibly one of the best choices since each segment or pad develops a pressurized oil wedge tending to center the shaft in the bearing, thereby increasing the system damping and overall stability.

6.083 Oil Whip Instability:

Oil Whip can occur on those machines subject to oil whirl when the oil whirl frequency coincides with and becomes locked into a system natural frequency (often a rotor "balance" or lateral natural frequency). For example, please refer again to Figure 6.08F. When the rotor speed increased to just above 9200 RPM, this brought its speed to 2X its first balance natural frequency. At this time, the oil whirl which was approximately 43% of RPM was brought into coincidence with this natural frequency. At this point, the oil whirl was suddenly replaced by oil whip- a lateral forward precessional subharmonic vibration of the rotor. At this point, the oil whip frequency remains the same, independent of the rotor RPM. Note that the oil whip frequency never changed even though the machine continued on up to 12,000 RPM. When a shaft goes into oil whip, its dominant dynamic factors become mass and stiffness in particular and its amplitude is limited only by the bearing clearance. Left uncorrected, oil whip may cause destructive vibration which can cause catastrophic failure.

6.084 Dry Whip:

Dry Whip occurs in journal bearing machines subject either to a lack of lubrication or the use of the wrong lubricant. When this occurs, excessive friction is generated between the stationary bearing and rotating journal. This friction can excite vibration in the bearing and other components. This kind of vibration is called Dry Whip. Dry Whip can also be caused by journal bearings having excessive clearance as well as those having insufficient clearance.

The Dry Whip condition is similar to taking a moistened finger and rubbing it over a dry pane of glass. It will generate a frequency specifically dependent upon the shaft and construction materials, geometries and lubricant properties. Normally, this frequency will be quite high producing something similar to a squealing noise similar to that produced by dry rolling element bearings. The frequency content itself will not be an integer multiple of the machine speed. When Dry Whip is expected, it is important that it be taken care of quickly in order to prevent potentially catastrophic failure. When Dry Whip is suspected, both the lubricant itself and the lubrication system should be closely inspected and the bearings should be checked to ensure that they have proper clearances.

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6.09 TRACKING OF ROLLING ELEMENT BEARING FAILURE STAGES USING VIBRATION AND HIGH FREQUENCY ENVELOPING AND DEMODULATED SPECTRAL TECHNIQUES

ABSTRACT:

Throughout the last decade, much research along with in-plant testing has been conducted in an attempt to accurately evaluate the condition of rolling element bearings. This has included studies to approximate remaining bearing life. This life is very dependent upon the vibration to which the bearing is subjected. The following paper will present a number of failure scenarios which have been identified to date for tracking rolling element bearing failure stages using both vibration and spike energy spectral analysis. These results have been directly correlated with studies which have been rigorously conducted in laboratories. Such studies have included evaluation of many types of rolling element bearings including deep groove ball, angular contact ball, needle, cylindrical roller, spherical roller and tapered roller bearings. It is the expressed purpose of this paper to provide the reader with solid tools with which he can not only evaluate the current health of specific rolling element bearings, but also can assist him in predicting remaining life and/or taking proactive steps immediately required to noticeably extend the life of the bearings.

INTRODUCTION:

A tremendous cross section of today's process and utility machinery is outfitted with rolling element bearings. In most all cases, these bearings are the most precise components within the machine, generally held to tolerances only 1/10th those of many of the remaining machine components. Yet, only about 10 to 20% of bearings achieve their design life due to a variety of factors. These primarily include lubrication inadequacies, use of the wrong lubricant, contamination with dirt and other foreign particles, improper storage outside their shipping packages, introduction of moisture, false brinelling during shipment or when standing idle, misapplication of the wrong bearing for the job, improper installation of bearings, etc.

Figure 6.09A shows the components of a rolling element bearing. Note the location of the accelerometer relative to the bearing's outer race, inner race, rolling elements and cage. When rolling element bearings wear, the vibration signal most readily travels from defects on the outer race to the accelerometer. Such flaws will normally appear on two or more of these components prior to eventual failure. Figure 6.09B illustrates various types of rolling element bearings.

One of the leading contributors to premature rolling element bearing failure is excessive vibration and the high dynamic loads that it can transmit into bearings. Following below is the design formula used in calculating theoretical ball bearing life which will show why it is so critical to bearing life to minimize the dynamic loads imposed upon them from vibration:

$$L_{10}\ \text{Life} = \left(\frac{16{,}666}{\text{RPM}}\right)\left(\frac{\text{RATING}_B}{\text{LOAD}_E}\right)^3 \quad \text{HOURS}$$

where:

L_{10} Life	=	No. of Hours that 90% of a group of bearings should attain or exceed prior to onset of fatigue failure.
RATING_B	=	Basic Dynamic Load Rating for a given bearing (lb)
LOAD_E	=	Equivalent Radial Load taken by a bearing - including Radial and Axial Loads (lb)

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FIGURE 6.09A
ROLLING ELEMENT BEARING TERMINOLOGY

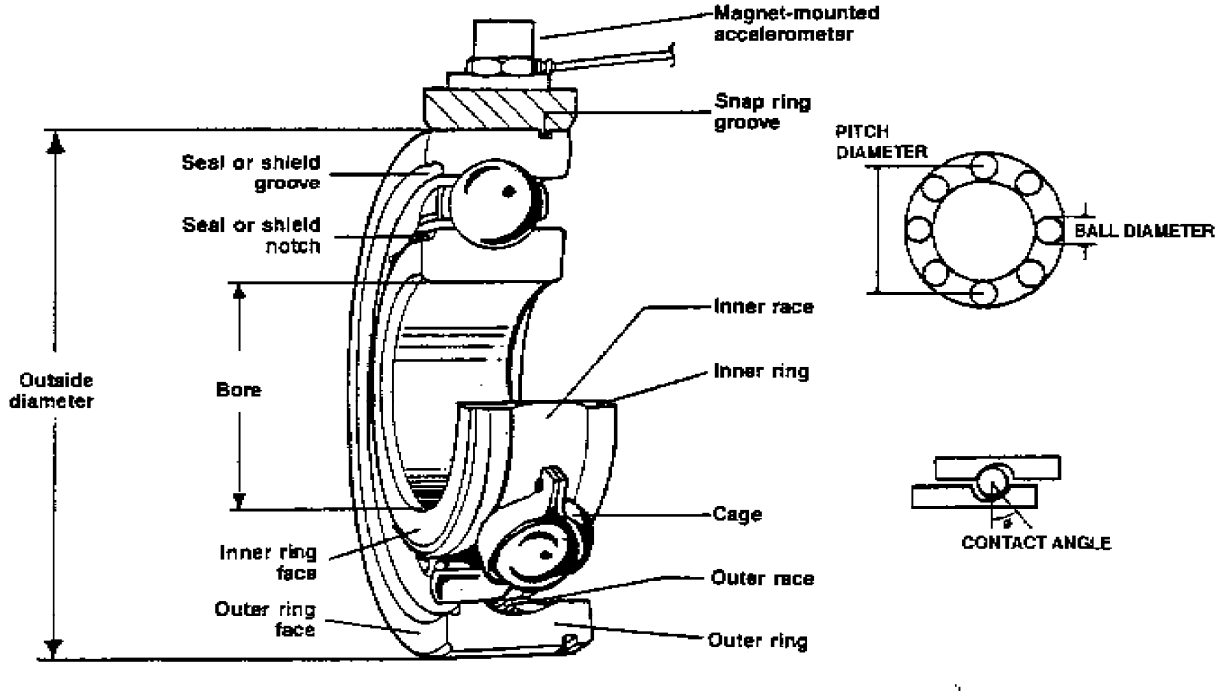
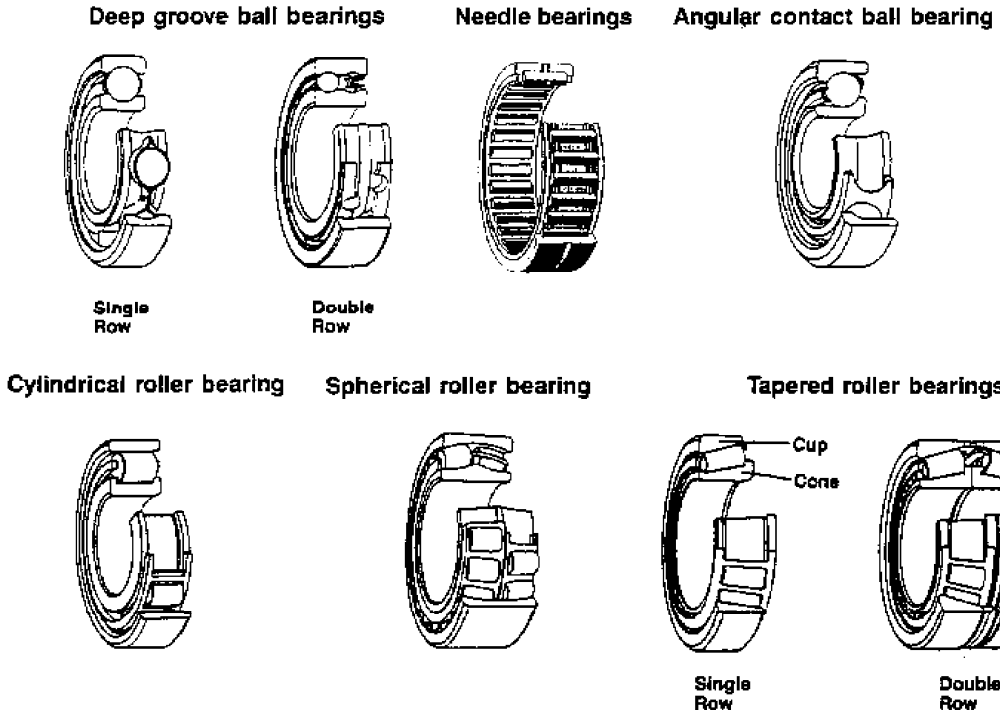


FIGURE 6.09B
ROLLING ELEMENT BEARING TYPES



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This formula shows that the greater the speed, the lower the anticipated life. However, of even greater significance is that the theoretical ball bearing life varies with the 3rd power of the load to which it is subjected (and to the 3.33 power in the case of roller bearings). Therefore, if the designer only considered the static loads on the bearing as well as those from other components such as belt tension, he may be surprised to learn of the magnitude of the dynamic forces which can be introduced from vibration.

The key point is that rolling element bearings in truth rarely fail due to defective workmanship themselves. In most all cases, outside influences act on the bearing to bring about its premature failure. If outside influences such as unbalance, misalignment, belt drive problems, soft foot, inadequate lubrication and improper installation can be taken care of, the bearings themselves should have satisfactory life.

To bring home the point, if one considers only the load from unbalance, this alone can generate significant dynamic loads. For example, consider a 2000 lb rotor turning at 6000 RPM with 1 oz of unbalance on a 3 foot diameter (18" radius). The amount of centrifugal force from the unbalance alone can be calculated as follows:

$$F_C = \frac{mrw^2}{g_C} = \left(\frac{Wr}{(386)(16)}\right)\left(\frac{2\pi n}{60}\right)^2$$

$$F_C = .000001775\ Un^2 = .00002841\ Wrn^2$$

where:

F_C = Centrifugal Force (lb)
U = Unbalance of Rotating Part (oz-in)
W = Weight of Rotating Part (lb)
r = eccentricity of the rotor (in)
n = Rotating Speed (RPM)

Now substituting for the sample rotor with a 1 oz unbalance at an 18" radius (U = 18 oz-in) turning 6000 RPM,

$$F_C = (.000001775)(18\ \text{oz-in})(6000\ \text{RPM})^2$$

F_C = 1150 lbs (from centrifugal force due to unbalance alone)

That is, only a 1 oz unbalance on a 3 foot diameter wheel turning 6000 RPM would introduce a centrifugal force of 1150 lbs which would have to be supported by the bearings in addition to the 2000 lb static rotor weight. Therefore, if the designer had only anticipated supporting 2000 lbs by the bearings, but in fact had to withstand 3150 lbs, his design life calculation would be off by a factor of:

$$\text{Corrected } L_{10} \text{ Life} = (\text{Initial Life})\left(\frac{2000}{3150}\right)^3$$

Corrected L_{10} Life = .25 X Initial Life Calculation (only 25% of design life)

Therefore, the actual theoretical life would only be 25% of his initial design life if this wheel were subjected only to unbalance, not to mention other dynamic forces introduced from vibration due to misalignment, looseness, cavitation or any other problems.

Of great importance is the ability to "track" the condition of rolling element bearings and to know when they will need replacement, right from the beginning when initial baseline signatures are

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acquired. Much has been learned during the past decade on how vibration signature analysis can contribute to this capability. Following in Section III will be a separate illustrated discussion on how this can be accomplished using not only vibration signature analysis, but also some of the newer high frequency enveloping signal processing techniques as well. This discussion will follow sections on "Which vibration parameter should normally be used?" (Section 6.091) and "What types of vibration frequencies are generated by defective rolling element bearings?" (Section 6.092). Then, recommendations on when rolling element bearings should be replaced will follow in Section 6.095.

6.091 OPTIMUM VIBRATION PARAMETER FOR ROLLING ELEMENT BEARING CONDITION EVALUATION (ACCELERATION, VELOCITY OR DISPLACEMENT)?

Particularly during this last decade, we have learned that we certainly can no longer depend on overall vibration alone to accurately evaluate the health of rolling element bearings. Also, we have found we cannot depend only on the measurement of ultrasonic frequency broadband measurements (Spike Energy, HFD, Shock Pulse, etc.) which make measurements in approximately the 5000 Hz to 60,000 Hz region. What we have learned is that these ultrasonic measurements are only an indicator, not the indicator of bearing health. They likewise are most effectively used in conjunction with vibration signature analysis to best evaluate bearing condition.

Of course, when employing vibration signature analysis, either of 3 vibration parameters can be used - acceleration (g), velocity (in/sec) or displacement (mils). Figure 6.09C shows how each of these parameters vary with frequency in terms of severity. Following below are comments on the attributes of each of these vibration parameters when specifically evaluating rolling element bearing health:

A. Displacement - Unfortunately, displacement spectra miss a great deal of bearing health information. Since displacement is low frequency intensive, it tends to suppress or almost eliminate much of the spectral content available that indicates bearing defect problems. In fact, one of the real problems with employing displacement on low-speed machines less than 200 RPM is the fact that while the spectrum may successfully display 1X RPM, it most often will almost completely miss bearing frequencies until the problem is quite severe.

B. Acceleration - Unlike displacement, acceleration tends to overemphasize much of the high frequency content generated by the rolling element bearing defects. As a result, if one is not greatly familiar with working with it, acceleration spectra might "cry wolf" far too often. For plant programs trying to establish themselves, this can do great harm to credibility. Although acceleration itself is probably a better indicator in the very early stages of bearing problems, it quickly gives way to vibration velocity which more accurately and clearly tells the "true story" of current bearing health.

C. Velocity - Velocity spectra should be one of the best parameters for evaluating most bearing problems, even on low-speed machines (for example, even if the speed was only about 60 RPM, much of the bearing frequency content would be above 500 CPM). For common rotational speeds ranging from 1200 to 3600 RPM, most of the spectral vibration content containing bearing defect information will be below 2000 Hz (120,000 CPM). In general, depending on the type of transducer employed, velocity will remain somewhat "flat" in frequencies ranging from 600 CPM up to 120,000 CPM (10 Hz - 2000 Hz). This means that somewhat "equal weight" can be assessed to a bearing defect frequency occurring at either 6000 CPM or at 60,000 CPM, whereas this could not be done with acceleration or displacement which are highly frequency dependent (see Figure 6.09C). When rolling element bearing machines operate at speeds above approximately 10,000 RPM, acceleration would then likely be the best rolling element bearing health indicator.

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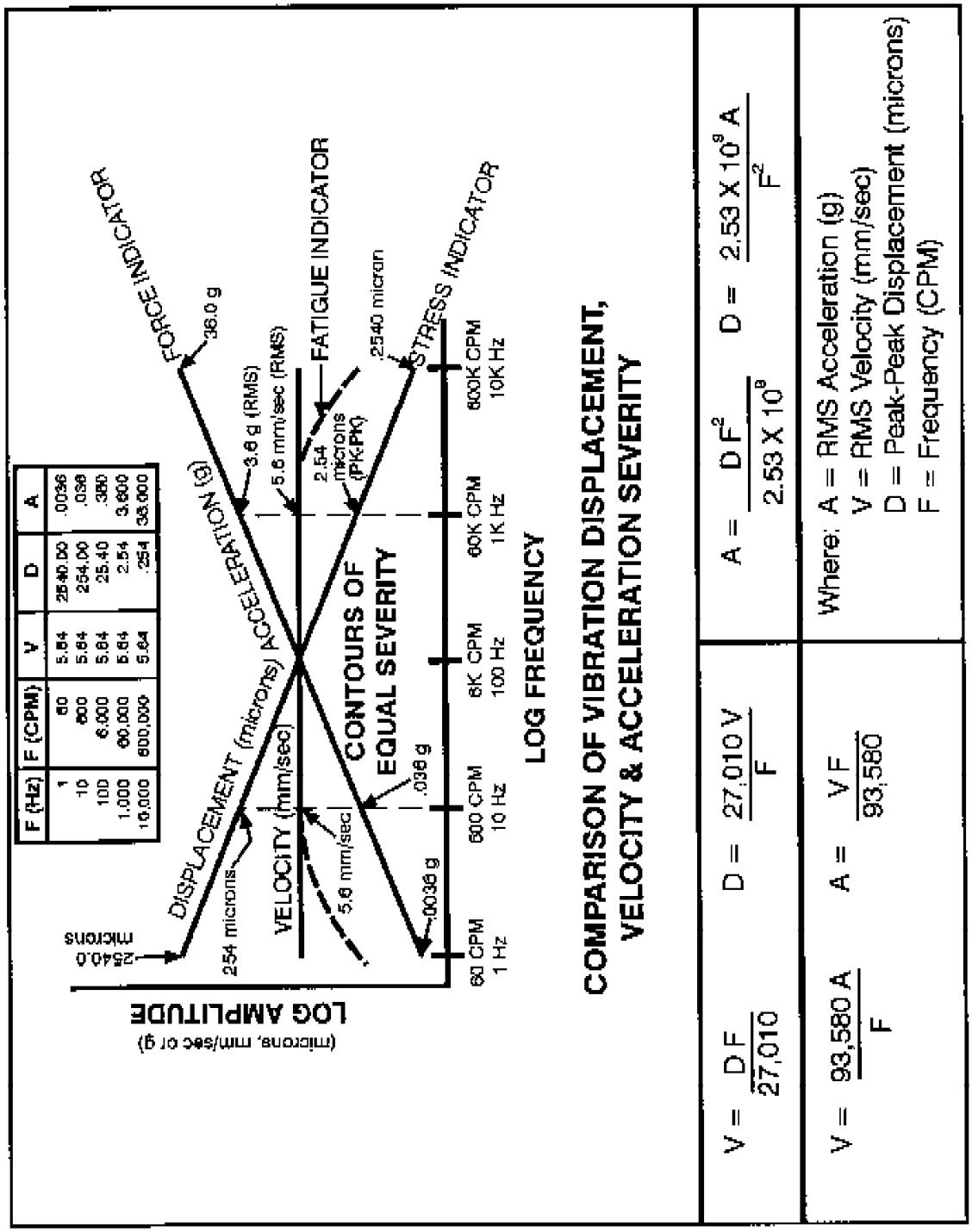
COMPARISON OF VIBRATION DISPLACEMENT, VELOCITY & ACCELERATION SEVERITY
LOG FREQUENCY
LOG AMPLITUDE
(microns, mm/sec or g)
CONTOURS OF EQUAL SEVERITY
DISPLACEMENT (microns)
VELOCITY (mm/sec)
ACCELERATION (g)
FORCE INDICATOR
FATIGUE INDICATOR
STRESS INDICATOR
25400 microns
254 microns
5.6 mm/sec
.036 g
60 CPM 1 Hz
600 CPM 10 Hz
6K CPM 100 Hz
60K CPM 1K Hz
600K CPM 10K Hz
36.0 g
3.6 g (RMS)
5.6 mm/sec (RMS)
2.54 microns (PK-PK)
.2540 micron
F (Hz) | F (CPM) | V | D | A
1 | 60 | 5.64 | 2540.00 | .0036
10 | 600 | 5.64 | 254.00 | .036
100 | 6,000 | 5.64 | 25.40 | .360
1,000 | 60,000 | 5.64 | 2.54 | 3.600
10,000 | 600,000 | 5.64 | .254 | 36.000
$V = \frac{DF}{27,010}$ $D = \frac{27,010\,V}{F}$ $A = \frac{DF^2}{2.53 \times 10^8}$ $D = \frac{2.53 \times 10^8\,A}{F^2}$
$V = \frac{93,580\,A}{F}$ $A = \frac{VF}{93,580}$
Where: A = RMS Acceleration (g)
V = RMS Velocity (mm/sec)
D = Peak-Peak Displacement (microns)
F = Frequency (CPM)

FIGURE 6.09C

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The remainder of this section on rolling element bearing health will assume velocity spectra. However, if one takes into account how the other parameters such as acceleration vary with frequency, he can apply many of these same techniques. Acceleration spectra may detect such bearing problems earlier than will velocity spectra, particularly on high-speed machinery. In addition, high frequency envelope demodulated spectra will likely provide even earlier warning of bearing wear and lubrication problems, and should be employed in addition to vibration spectra alone on critical machinery (this will be discussed in Section 6.093 of the paper).

6.092 TYPES OF VIBRATION SPECTRA CAUSED BY DEFECTIVE ROLLING ELEMENT BEARINGS:

Defective rolling element bearings generate each of 4 types of frequencies once they begin to develop defects. These frequencies include (a) random, ultrasonic frequencies; (b) natural frequencies of bearing components; (c) rotational defect frequencies; and (d) sum and difference frequencies. Following below will be a discussion on each of these types of frequencies and their significance:

(a) Random, Ultrasonic Frequencies:

Measurements in this ultrasonic frequency region ranging from approximately 5000 Hz to 60,000 Hz are made by a variety of instruments and employ a similar variety of techniques. These include spike energy, HFD high frequency acceleration, shock pulse measurement and others. Each of these are meant to be incipient failure detection parameters that can track bearing health from its installation until just prior to or including eventual failure. Each of them have their own strengths and weaknesses, not only concerning their bearing health evaluation accuracy, but also how well each of them can be trended with time. In general, the overall number they provide gives just one more piece of information to be considered when evaluating bearing health. However, the information contributed by the vibration spectral data should be given significantly more weight.

Figure 6.09D provides a severity chart for specifying alarm levels of Spike Energy. This chart shows that machine speed must be taken into account when evaluating Spike Energy amplitudes. Similarly, Figure 6.09E provides a comparable severity chart for “HFD” and Shock Pulse (SPM), comparing the relative amplitudes of these parameters with Spike Energy levels in a study conducted by Mr. Charles Berggren (References 16 & 19).

(b) Natural Frequencies of Bearing Components (when installed):

Reference 1 documents that the natural frequencies of installed rolling element bearing components range from approximately 500 to 2000 Hz (30,000 to 120,000 CPM). Like every other member, these bearing components “ring” at these natural frequencies when they are impacted. In the case of rolling element bearings, intermittent impacts of the rolling elements striking flaws on the raceways ring their natural frequencies. Actually, there are several bearing component natural frequencies in the region of 30,000 to 120,000 CPM, but some are much more predominant than others. Therefore, when defects progress beyond microscopic size, they begin to excite these natural frequencies making them the “second line of detection” (Failure Stage 2, discussed in Section 6.093). As the defects worsen, they can cause greater impacts which cause greater response from the natural frequency peaks. Eventually, when wear progresses, more frequencies around these resonances appear, many of which will be 1X RPM sidebands of these natural frequencies (often, such modulating peaks will be spaced at bearing defect frequencies rather than 1X RPM sidebands).

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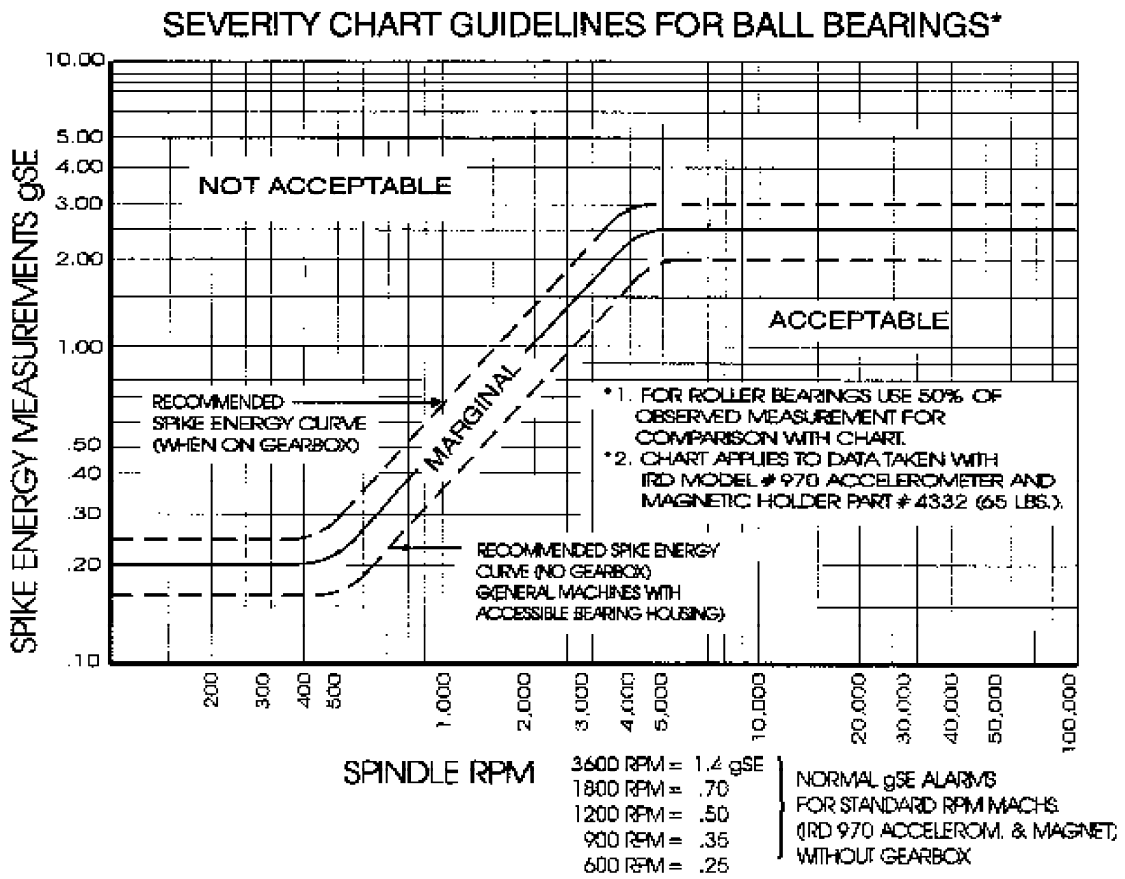
FIGURE 6.09D
RECOMMENDED SPIKE ENERGY SEVERITY CHART (IRD SPIKE ENERGY)



SPIKE ENERGY™ MEASUREMENTS

Energy is generated by repetitive transient mechanical impacts. Such impacts typically occur as a result of surface flaws in rolling-element bearings or gear teeth.

This energy is conducted from its source through various paths to the outer surface of the machine structure, and is seen as a small-amplitude vibration at the surface. Accelerometers coupled to the surface generate corresponding electrical signal.

The accelerometer signals processed by unique filtering and detection circuitry to produce a single "figure of merit" related to the intensity of the original impacts. This figure of merit is expressed in "gSE" units.

SPIKE ENERGY™ gSE readings are measurements which can with experience, be correlated with the severity of the casual surface flaws. Even though gSE readings are affected by the nature of the conductive path between the impact source and the accelerometer, similar machine structures will provide a reasonable basis for comparison between the structures.

The gSE figure of merit has proven to be effective in detecting mechanical defects in meshing gears and rolling element bearings. The gSE measurement, when used in conjunction with conventional measurement of vibration velocity and acceleration, provides early indications of mechanical deterioration.

** When used with magnetic holders, accelerometers must be installed with a light coating of silicone grease and tightened to 40 in-lb. torque.

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FIGURE 6.09E
MAINTENANCE DIAGNOSTIC VIBRATION AND HIGH FREQUENCY GENERAL TOLERANCE CHART FOR PROCESS MACHINERY WITH ROLLING ELEMENT BEARINGS①

Unfiltered Vibration Velocity	Shock Pulse & Acoustic Emission	Microlog HFD	IRD Spike Energy ②	Severity Level and Alarm Status
Inches/Second Peak	Db	g	gSE	
1.5+	50+	5.0+	3.0+	Danger (Shutdown)
.75 - 1.49	40 - 49	3.0 - 4.99	1.50 - 2.99	Very Rough (Alert)
.40 - .74	30 - 39	1.50 - 2.99	.80 - 1.49	Rough (Alert)
.20 - .39	20 - 29	.75 - 1.49	.40 - .79	Fair (Acceptable)
.10 - .19	10 - 19	.30 - .74	.20 - .39	Good (Acceptable)
.01 - .09	1 - 9	.01 - .29	.01 - .19	Smooth (Acceptable)

① Provide by Charles Berggren of Monsanto
② Spike Energy Amplitudes measured using an IRD 970 accelerometer outfitted with IRD's 2-pole, 65 lb. magnet.

One important point about these bearing component natural frequencies is the fact that they are independent of running speed. That is, whether the shaft is turning at low or at very high speeds, the natural frequencies remain at the same frequency location. However, their response amplitude is proportional to the impact velocity, which means that with greater rotational speeds, they normally will respond at greater amplitudes.

(c) Rotational Defect Frequencies:

Through the years, a series of formulas have been developed which can help detect specific defects within rolling element bearings. They can separately detect faults on the inner race, outer race, cage or rolling elements themselves. They are based on the bearing geometry, the number of rolling elements and the bearing rotational speed. Figure 6.09F provides the formulas for each of these four rolling element bearing defect frequencies.

The power of these equations is that if one knows the design parameters of his bearings (pitch diameter, rolling element diameter, number of rolling elements & contact angle), he is able to detect problems which occur on the races, cage or rolling elements, and he is enabled to track these problems as deterioration continues. In many cases, the analyst may not know all the parameters to insert for a particular bearing in the equations, but he might know the bearing manufacturer and model number. In these cases, there are several publications and software offerings which tabulate each of the 4 defect frequencies for each of the bearing model numbers. Figure 6.09G is an example of one of the better known publications listing these defect frequencies (as per Reference 12). Note on Figure 6.09G that this sheet provides the number of rolling elements (N_b), rolling element diameter (B_d), bearing pitch diameter (P_d), contact angle

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(∅), outer race defect frequency (BPFO), inner race defect frequency (BPFI), cage defect frequency (FTF) and rolling element defect frequency (BSF). Note that Figure 6.09G also provides an example assuming the SKF N220 bearing. Note that each one of the bearing frequencies are given in terms of running speed orders. Therefore, the frequencies for the SKF N220 bearing show, for example, that the inner race frequency (BPFI) is 9.13X RPM. Thus, if this bearing were turning at 1000 RPM, and a frequency was detected at 9130 CPM, he could conclude that this peak is the bearing inner race defect frequency and that a fault is developing there. On the other hand, if a frequency of 6860 CPM occurred on this same unit, he would know this is the outer race defect frequency of this SKF N220 bearing (since BPFO = 6.86X RPM).

The great advantage of knowing these bearing defect frequencies is that, for example, an analyst can separately evaluate the outboard and inboard bearings on the same machine, particularly if they are different model numbers. For example, assume that the outboard bearing of our example was an SKF N220 and that its inboard bearing was an SKF N228 bearing. Note from Figure 6.09G that they would have an entirely different set of defect frequencies (for example, the inner race frequency on the N228 would be 10.19X RPM as compared to 9.13X RPM on the N220).

A number of interesting facts can be stated about these bearing defect frequencies as follows:

1. How Bearing Frequencies Differ From Other Frequencies (Defect Frequencies):

One thing setting rolling element bearing defect frequencies apart from other vibration sources is the fact that they are defect frequencies. In other words, bearing defect frequencies should not be present. When they are present, they signal at least an incipient problem. On the other hand, other common frequencies such as 1X RPM are always present whether or not there is satisfactory or unsatisfactory balance or alignment; pumps and vanes always show vibration at some amplitude for blade pass frequencies; gears cause vibration at the number of teeth X RPM. However, the presence of these other frequencies does not mean there is necessarily a defect or problem. The appearance of bearing defect frequencies sends a message to the analyst to "pay attention". However, it is also important to point out the presence of such defect frequencies do not necessarily mean there are defects within the bearing. They also will appear if there is insufficient lubricant allowing metal-to-metal contact, or if the bearings are improperly loaded (excessive press fit, excessive thrust on a bearing not necessarily designed to take thrust, or if a thrust bearing is installed backwards, etc.).

2. Bearing Defect Frequencies are Noninteger Multiples of Operating Speed:

Referring back to Figure 6.09G, note that each of the bearing defect frequencies are noninteger multiples. That is, they are one of the few machinery vibration sources that do not generate integer multiples of rotational speed (GMF = #teeth X RPM; BPF = #blades X RPM). This is helpful when the manufacturer and model number of the bearing is unknown. Look for real number RPM multiples (such as 5.78X, or 7.14X, etc.).

3. Sum of Race Frequencies = "Bearing Ball Pass Frequency":

Figure 6.09F shows an interesting relationship between the outer and inner race frequency multipliers and the number of rolling elements. Note that the product of the number of balls X RPM equals the sum of the outer and inner race frequencies (BPFO + BPFI). For years, many have looked for the appearance of frequencies at the number of balls X RPM similar to what they have experienced with gears (#teeth X RPM), blade pass frequency (#Blades X RPM) and so forth. However, the author has rarely seen the appearance of a frequency at "bearing ball pass frequency", but has commonly seen each of the race frequencies themselves which are on the other side of the equation.

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FIGURE 6.09F
ROLLING ELEMENT BEARING DEFECT FREQUENCIES

These equations assume the inner race is rotating with the shaft while the outer race is fixed (stationary):

INNER RACE = BPFI =
$$\frac{N_b}{2}\left(1+\frac{B_d}{P_d}\cos\theta\right) X\ RPM$$

OUTER RACE = BPFO =
$$\frac{N_b}{2}\left(1-\frac{B_d}{P_d}\cos\theta\right) X\ RPM = N_b\ X\ FTF$$

BALL (OR ROLLER) = BSF =
$$\frac{P_d}{2B_d}\left[1-\left(\frac{B_d}{P_d}\cos\theta\right)^2\right] X\ RPM$$

CAGE = FTF =
$$\frac{1}{2}\left(1-\frac{B_d}{P_d}\cos\theta\right) X\ RPM = .35 - .45X\ RPM$$

NOTE:
If the inner ring is fixed (stationary) while the outer race rotates with the shaft, the minus sign must be changed to a plus sign within the parentheses of the Cage Frequency Equation. In this case, N_b X FTF will now equal BPFI rather than BPFO; and FTF = .55 - .65X RPM. However, BPFO, BPFI and BSF will remain the same.

(NOTE THAT BPFI + BPFO = N_b)

where:

N_b = Number of Balls or Rollers

B_d = Ball or Roller Diameter (in. or mm)

P_d = Bearing Pitch Diameter (in. or mm)

θ = Contact Angle (degrees)

APPROXIMATIONS (If Bearing Dimensions are Unknown, but do know # of Balls or Rollers) (Reference 18):

Approx. BPFI ≈
$$\left(\frac{N_b}{2}+1.2\right) X\ RPM$$

Approx. BPFO ≈
$$\left(\frac{N_b}{2}-1.2\right) X\ RPM$$

Approx. BSF ≈
$$\frac{1}{2}\left(\frac{N_b}{2}-\frac{1.2}{N_b}\right) X\ RPM$$

Approx. FTF ≈
$$\left(\frac{1}{2}-\frac{1.2}{N_b}\right) X\ RPM$$

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FIGURE 6.09G
EXAMPLE TABULATION OF ROLLING ELEMENT BEARING DEFECT FREQUENCIES (REFERENCE 12)

SKF

SINGLE ROW CYLINDRICAL ROLLER BEARINGS

BEARING NUMBER	Nb	Bd	Pd	Ø	BPFO	BPFI	FTF	BSF
N205	12	.256	1.516	0	4.99	7.01	.41	2.87
N206	12	.295	1.811	0	5.00	6.94	.41	2.99
N207	12	.354	2.106	0	4.97	6.98	.41	2.89
N208	13	.394	2.362	0	5.42	7.58	.41	2.92
N209	14	.394	2.559	0	5.97	8.07	.42	3.17
N210	15	.394	2.968	0	6.50	8.48	.43	3.70
N211	15	.433	3.051	0	6.43	8.56	.43	3.45
N212	16	.472	3.476	0	6.92	9.08	.43	3.61
N213	16	.512	3.641	0	6.88	9.11	.43	3.49
N214	16	.512	3.738	0	6.93	9.06	.43	3.58
N215	17	.551	4.035	0	7.54	9.65	.43	3.59
N216	16	.590	4.331	0	6.91	9.08	.43	3.60
N217	16	.620	4.626	0	6.91	9.08	.43	3.60
N218	16	.709	4.921	0	6.84	9.15	.43	3.40
N219	16	.748	5.216	0	6.85	9.14	.43	3.41
N220	16	.787	5.512	0	6.86	9.13	.43	3.43
N222	16	.905	6.102	0	6.81	9.18	.42	3.59
N224	16	.945	6.594	0	6.85	9.14	.42	3.41
N225	18	.945	7.087	0	7.91	10.07	.43	3.32
N228	18	1.024	7.677	0	7.76	10.19	.43	3.68
N305	10	.354	1.713	0	4.01	5.99	.40	2.32
N306	11	.394	2.007	0	4.42	6.58	.40	2.45
N307	11	.433	2.264	0	4.45	6.55	.40	2.51
N308	12	.472	2.559	0	4.88	7.07	.41	2.62
N309	11	.551	2.854	0	4.43	6.56	.40	2.49
N310	12	.590	3.150	0	4.85	7.09	.40	2.57
N311	11	.669	3.445	0	4.42	6.58	.40	2.48
N312	12	.709	3.740	0	4.85	7.10	.41	2.54
N313	12	.748	4.035	0	4.87	7.08	.41	2.60
N314	12	.787	4.330	0	4.89	7.06	.41	2.66
N315	12	.866	4.626	0	4.85	7.09	.41	2.57
N316	13	.866	4.921	0	5.36	7.63	.41	2.75
N317	12	.945	5.216	0	4.89	7.06	.41	2.67
N318	13	.984	5.512	0	5.34	7.65	.41	2.71
N319	14	1.024	5.807	0	5.77	8.22	.41	2.74
N320	14	1.102	6.201	0	5.75	8.24	.41	2.68
N321	14	1.181	6.496	0	5.72	8.27	.41	2.66
N322	14	1.260	6.890	0	5.72	8.27	.41	2.64

EXAMPLE:
FOR SKF N220 BEARING, BPFI = 9.13 X RPM (Inner Race)
BPFO = 6.86 X RPM (Outer Race)
BSF = 3.43 X RPM (Rolling Elements)
FTF = .43 X RPM (Cage Freq.)

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4. Description of How Defect Frequencies are Generated Within Rolling Element Bearings:

Figure 6.09F provides the formulas required to calculate the four defect frequencies for each rolling element bearing (BPFI, BPFO, BSF and FTF). Figure 6.09H illustrates how such defect frequencies are generated within the bearings. For example, if there is a defect on the outer race on the bottom of the bearing within the load zone as shown in Figure 6.09H, note that an impulse occurs in the time waveform at each instant when a rolling element passes over and impacts this defect. On the other hand, if the inner race had a defect, an impulse would occur in the time domain when the inner race rotated past each rolling element (assuming the inner race is press fit on the shaft). An important fact shown by Figure 6.09H is that the amount of response from the rolling elements striking the inner race defect will depend on where the inner race is positioned at that particular instant of time when the impact occurs (that is, if the inner race defect is positioned within the load zone, it will have significantly more response than it would if the impact occurred with the same inner race defect positioned 180° away, clearly out of the load zone). This explains why inner race defect frequencies are often surrounded by sidebands spaced at 1X RPM since their amplitude is modulated at the rate of once per revolution as shown by Figure 6.09H. On the other hand, since the outer race does not rotate in this instance, the amplitude response in the time domain should remain near constant. Therefore, 1X RPM sidebands surrounding outer race frequencies (BPFO) are much more serious than those surrounding the inner race frequency (BPFI), again assuming the inner race is press fit on the shaft (if the inner race is stationary while the outer race rotates, the reverse with respect to sidebands would occur). The presence of 1X RPM sidebands surrounding outer race (BPFO) frequencies normally means the problem is sufficiently serious to cause the bearing to actually impede the motion of the shaft. As Section 6.093 will later point out, when the inner race frequency (BPFI) becomes surrounded by several "families" of 1X RPM sidebands, this can likewise indicate a more serious problem.

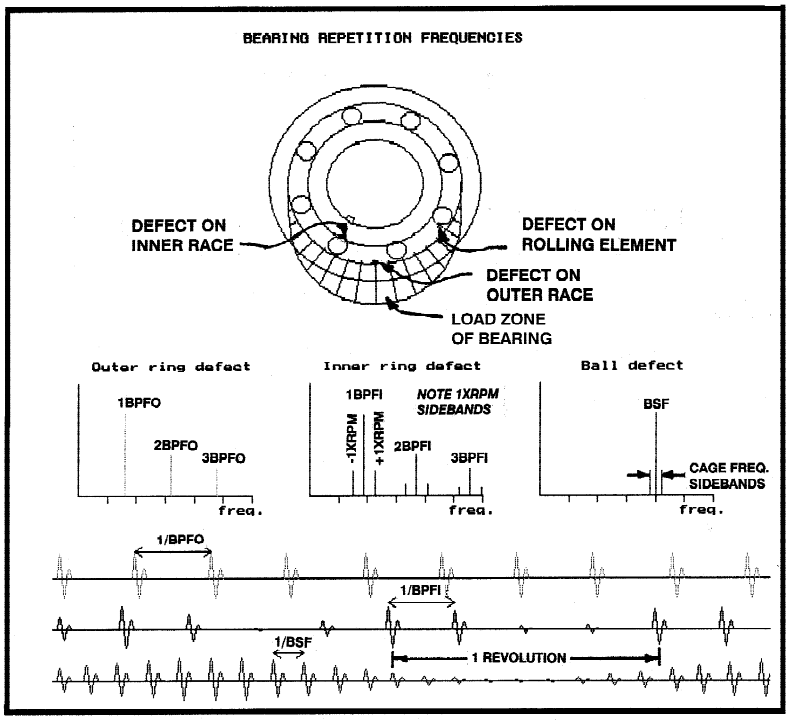

FIGURE 6.09H
ILLUSTRATION OF HOW DEFECT FREQUENCIES ARE GENERATED WITH ROLLING ELEMENT BEARINGS

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5. Relative Outer and Inner Race Amplitudes:

Normally, the amplitude of the outer race frequency is higher than that of the inner race frequency. This is likely due to the fact that the transducer itself is much closer to the outer race (see Figure 6.09A). In addition, the vibration signal from the inner race frequency (or harmonics) must pass through several more interfaces including the constantly rotating rolling elements on its path out to the vibration transducer.

6. Usual Order of Appearance by Defect Frequencies:

Normally, defects will first appear on the races before the problem propagates to the rolling elements and cage. Thus, outer and inner race frequencies are normally first established before the appearance of a ball spin frequency. Later, the cage frequency normally can appear either as a fundamental or as a sideband to another frequency. Likewise, the ball spin frequency will sometimes appear as a sideband above and below an outer or inner race defect frequency.

7. Where Cage Frequency Normally Appears:

Although rolling element bearing problems are classically thought of as high frequency problems, fundamental cage frequencies will always be subsynchronous, ranging from approximately .33X RPM up to .48X RPM with the majority falling between .35X and .45X RPM. However, the cage frequency will not normally appear at its fundamental frequency. Instead, it most often will appear as a sideband around ball spin frequency (BSF), or around one of the race frequencies (BPFO or BPFI) with the sideband difference frequency equal to the cage frequency (FTF). For example, see Figure 6.09I which shows an outer race frequency BPFO) at 10,260 CPM (5.78X RPM) with cage frequency sidebands (FTF) spaced at 720 CPM (.41 X running speed which is 1775 RPM). This is how a cage frequency most often will appear.

8. Frequencies Generated by Faults on Balls or Rollers:

When defects occur on rolling elements themselves, they will often generate a frequency not only at the ball spin frequency (BSF), but also at the cage frequency (FTF, also known as fundamental train frequency).

9. Ball Spin Frequency May Appear if Cage is Broken:

The appearance of a ball spin frequency does not always necessarily mean there is a defect on the rolling elements. However, it still means that there is a problem present. In this case, it can indicate that a cage is broken at a rivet and if the balls are thrusting hard against the cage (as per Reference 12).

10. Frequency Generated if More than One Rolling Element has Faults:

If more than one rolling element has defects, a frequency equal to the number of balls having defects X the ball spin frequency will be generated. In other words, if defects are present on 5 balls or rollers, a frequency at 5X Ball Spin Frequency would most often appear.

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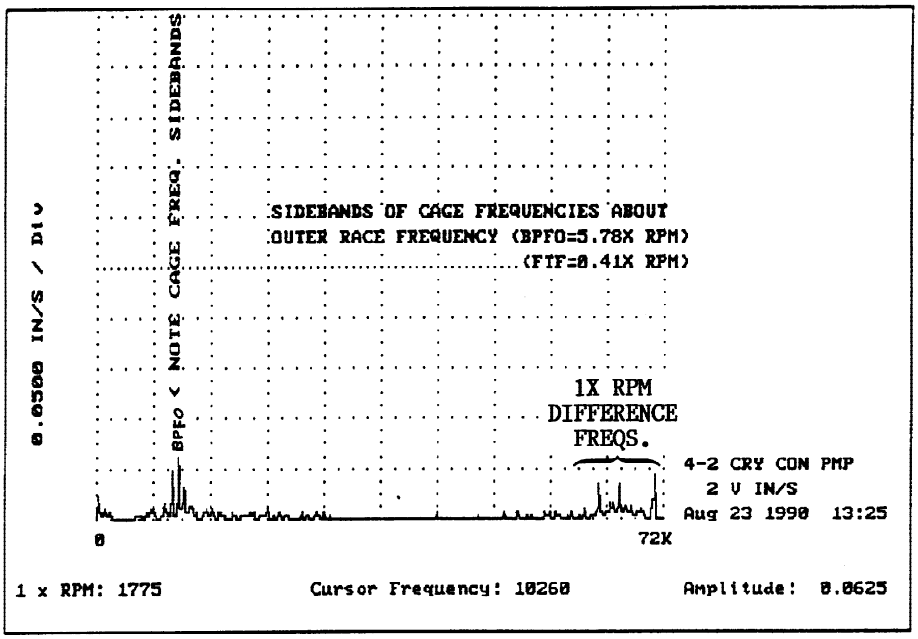

002% x 0.500 = 0.0100 1 x RPM: 1775.00

Machine	Pos	Dir	Units	Date
4-2 CRY CON PMP	2	V	IN/S	Aug 23 1990 13:25

Frequency	1 x RPM	Amplitude		Frequency	1 x RPM	Amplitude
180	0.10	0.0225		67140	37.83	0.0125
1080	0.61	0.0100		67860	38.23	0.0125
6840	3.85	0.0100		69120	38.94	0.0100
7200	4.06	0.0125		70920	39.95	0.0425
8640	4.87	0.0150				
9540	5.37	0.0475 ← -FTF (.41x)				
10260	5.78	0.0625 ← BPFO				
10980	6.19	0.0325 ← +FTF (.41x)				
11880	6.69	0.0125				
13860	7.81	0.0100				
14580	8.21	0.0100				
21600	12.17	0.0100				
61920	34.88	0.0100				
63720	35.90	0.0350				
65160	36.71	0.0150				
65520	36.91	0.0150				
66420	37.42	0.0350				

FIGURE 6.09I
HOW THE CAGE FREQUENCY (FTF) MOST OFTEN APPEARS WITHIN A VIBRATION SPECTRUM

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11. Allowable Vibration at Bearing Defect Frequencies:

It is very difficult to assign definite vibration severity levels to bearing defect frequencies similar to that which is commonly done for unbalance amplitude at 1X RPM. For one, there are a variety of rolling element bearings in a variety of different machines, each of which may provide different paths for the vibration signal to the transducer. However, one can make a statement which will apply almost uniformly for all the various types and combinations - the allowable vibration for unbalance at 1X RPM is much greater than that allowed for a defect frequency for a rolling element bearing.

Much research has proven that no absolute answer can be given to allowable vibration amplitudes at bearing defect frequencies. Not only does it depend on which particular machine a bearing is installed and what the operating speed is, it also depends greatly on which bearing failure scenario path it will travel. Each of 6 bearing failure scenarios are identified in Section 6.093. For example, on rare occasions, the author has personally witnessed bearings which still did not have significant damage even with an amplitude of .30 in/sec at a fundamental BPFO when this damage was concentrated at one particular location on the outer race (as described by Bearing Failure Scenario B in Section 6.093). On the other hand, considerable damage has been discovered in other bearings when no one defect frequency had an amplitude greater than only .03 in/sec in everyday machinery like pumps and blowers running at common speeds such as 1780 RPM. In fact, extensive damage has been found in large dryer roll bearings on paper machines running less than 100 RPM with bearing frequency amplitudes that ranged from only .003 to .006 in/sec.

The key point common to each of the latter two scenarios having low vibration but significant bearing damage was that in each case, not just one bearing frequency was present in the spectra; instead, a number of bearing frequency harmonics were present (a "number of bearing frequencies present" means for example that either 3 to 5 BPFO or BPFI harmonics might simultaneously be present; or that 2 or more harmonics of BPFO might be present along with 2 or more harmonics of BPFI). In addition, further investigation has shown that when these bearing frequency harmonics were surrounded by sideband frequencies spaced at 1X RPM of the problem bearing, even more damage is indicated (particularly if these 1X RPM sidebands surround BPFO harmonics, assuming the bearing is press fit on the shaft). It should also be emphasized that these sidebands may be spaced at bearing frequencies themselves rather than at 1X RPM (i.e., 4 or 5 BPFO harmonics may all be present, each with sidebands of FTF or BSF above and below them which would likewise indicate a potentially serious problem).

Therefore, the most important thing to look for indicating significant bearing wear is the presence of a number of bearing defect frequency harmonics, particularly if they are surrounded by sidebands spaced at either 1X RPM or sidebands spaced at other defect frequencies of the bearing - independent of amplitude. If these are present in a spectrum, replace the bearing as soon as possible.

12. Evaluating Bearings on Low-Speed Machinery (less than 250 RPM):

Reference 9 points out special precautions must be taken when making measurements on low-speed machinery. Rolling element bearings have been successfully evaluated at speeds as low as 1.5 RPM. However, one must be particularly aware of the low frequency limitations of both his analyzer and his transducer. If he uses one of the computer-loaded data collectors popular today, he should be aware that many of these instruments are outfitted with high-pass filters that begin filtering out signals below fixed frequency ranges at rates of from approximately 12 to 24 dB/ octave. This cutoff frequency may be close to DC (0 Hz) on some data collectors, but may be as high as 8 Hz (480 CPM) on other models. Thus, in the

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latter case, a machine may be generating 1.00 in/sec of vibration at 240 RPM, but the instrument may display only approximately .25 in/sec at this frequency if it has such a high-pass filter. Obviously, if one attempted to make measurements at frequencies below 120 CPM with this instrument, the data collector would detect and display only a small percentage of this peak. And, if one wanted to single or double integrate the signal from acceleration to velocity or displacement, respectively, there would likely be even greater suppression of the amplitude in the data collector (also, it is important to point out that some data collectors do not integrate acceleration signals to velocity at frequencies below approximately 120 CPM; data below this cutoff frequency is left nonintegrated). In addition to filtering, the low frequency response of the analyzer is also effected by its signal conditioning and input circuitry components which also needs to be considered.

Not only does one have to be concerned about the analyzer during low frequency measurements, he also must be aware of the frequency response of the transducer whether it is an accelerometer, a velocity pickup, or a proximity probe. Typically, most general purpose accelerometers in use with data collectors today are flat within 5% between approximately 5 Hz and 10,000 Hz (300 - 600,000 CPM). However, special seismic piezoelectric accelerometers can be obtained which will extend the ±5% flat response down to as low as approximately 0.1 Hz (6 CPM). These transducers will typically have exceptionally high sensitivities ranging from 1000 to 10,000 mV/g (as compared to only 10 to 100 mV/g for the standard accelerometer) and will normally have much greater weight on the order of 400 to 1000 grams (as compared to only 10 to 50 grams for the standard unit). They also will typically have much longer discharge time constants of 20 seconds or more (compared to only 0.5 second for standard accelerometers) and will work best when connected to analyzers having input impedances of approximately 1,000,000 ohms. Finally, the low frequency performance of these transducers can be further enhanced by connecting them to special power supplies and signal conditioning equipment.

In addition to the instrument limitations discussed, it should be pointed out that both 1X RPM and bearing frequency amplitudes themselves will inherently be much lower on large, low-speed machinery. In the case of 1X RPM, there will be minimal unbalance forces since these vary with the square of speed. Thus, if one attempts to specify spectral alarm bands for this machinery, he will have to spec them at much lower alarm amplitudes than for machinery rotating above 1200 RPM. The best way to specify them will be to capture actual data and perform statistical analysis on the overall levels and those of individual frequency bands.

Fortunately, even though many of today's standard data collectors and transducers may not be capable of evaluating 1X RPM and 2X RPM vibration on much low-speed machinery, these same instruments might still be able to satisfactorily evaluate the health of their bearings. Please refer to Figure 6.09G which shows some typical rolling element bearing frequencies. Even though the fundamental cage (FTF) and ball spin frequencies (BSF) may still be below the "reach" of the system, the more common outer race (BPFO) and inner race frequencies (BPFI) will most often range from 4 to 12X RPM. Therefore, a measurement system which could not "see" vibration at, for example, 100 RPM speed very likely could detect fundamental outer and inner race frequencies which would probably range from about 500 to 1000 CPM on this machine with little or no loss of signals. And, of course, harmonics of these bearing frequencies could easily be evaluated.

Importantly, when evaluating bearings on machines particularly running below 100 RPM, it is highly recommended that both time waveform and FFT spectra be captured. Often, when speeds drop to these levels, impulses which occur when the rolling elements roll past defects on the races will not have sufficient energy to generate clearly detectable defect frequencies in FFT spectra, but still might clearly be seen in the time waveform. One such example is given in Figure 6.09J in which case the time waveform is shown directly below the FFT

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spectrum. In this particular case, the analyst would likely have seen the inner race defect frequencies present in the FFT spectrum. However, note they were likewise clearly present at much greater amplitude in the time waveform. For example, the time waveform amplitude of these pulses spaced at the inverse of the inner race frequency (1/BPFI) had amplitudes of about .25 in/sec, compared to amplitudes of about only .01 in/sec in the FFT spectrum (or about 25 times higher amplitude in the waveform). This is very common with defects associated with rolling element bearings and gears which cause short-lived impacts in the waveform. When the FFT mathematics are performed, these waveform transient pulse amplitudes are often heavily suppressed when converted into the spectrum, and often will not even be noticed, particularly if there are frequencies present associated with other mechanical problems of much higher amplitude (such as unbalance at 1X RPM, misalignment at 2X RPM, blade pass frequency, etc.).

When speeds drop to very low levels on the order of 1 to 20 RPM, FFT spectra alone will almost never detect rolling element bearing problems. However, the time waveform may still reveal the bearing problems as demonstrated by Figure 6.09K which from Reference 13. In this example, taken on a 200 ton BOF (basic oxygen furnace) vessel trunnion bearing in a steel mill turning at a nominal speed of 1.5 RPM (40 sec/rev), the time waveform clearly showed an outer race defect frequency (BPFO) when the spectrum did not. Here, BPFO equaled 20.4 CPM (0.34 Hz) as compared to the operating speed of 1.5 RPM (0.025 Hz). Notice each of the pronounced spikes produced when the trunnion bearing rollers passed over and impacted the defects on the outer race (also note that the machine speed actually varied from about 1.3 to 1.7 RPM during the analysis). The frequency was calculated by measuring the time between impacts (sec/cycle) and inverting the result. For example, referring to Figure 6.09K, taking the first annotated times (3.367 sec - 0.427 sec = 2.940 sec), these would correspond to a frequency of 1/2.940 sec or 0.34 Hz (20.4 CPM), which equalled BPFO for this bearing.

Reference 26 points out that when trying to decide upon an optimum sampling time (t_{MAX}) which should be specified for low-speed machines, one should normally specify a sampling time equaling approximately 2 to 4 shaft revolutions. That is, if the machine were rotating at 60 RPM (1.0 rev/sec), this would correspond to 1 sec/rev. Therefore, a t_{MAX} of about 2 to 4 seconds should be specified to capture such bearing problems, in addition to capturing the FFT spectrum as well. On the other hand, if the machine were running at only 10 RPM (6 sec/rev), a sampling time (t_{MAX}) of about 12 to 24 seconds should be employed. It should be pointed out for very long sampling times such as these, it would be better to specify at least 2048 samples (corresponding to an 800 line FFT) rather than the usual 1024 samples (400 line FFT). Doing so will provide a sufficient number of samples so that the full amplitudes will be traversed in the time waveform, besides drawing a much truer picture of the actual vibration response of the machine in the waveform. Note that such time waveforms should be non-synchronous waveforms rather than synchronous time waveforms (since synchronous waveforms actually remove bearing defect frequencies from both the waveform and FFT since they are not integer multiples of running speed).

For now, the most important point is that one can successfully evaluate bearing health on low-speed machinery if he takes into account the frequency response performance characteristics of his instrumentation system and takes the necessary provisions outlined in this section.

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RECOMMENDED SAMPLING TIME (t_{MAX}) AND NUMBER OF SAMPLES
FOR TIME WAVEFORMS ON VARIOUS LOW-SPEED MACHINES

RPM	SEC / REV	RECOM. t_{MAX}	RECOM. # SAMPLES	RESULTANT F_{MAX}
600	.100	.400 sec	1024	1000 Hz (60,000 CPM)
500	.120	.400		1000 Hz (60,000 CPM)
450	.133	.400		1000 Hz (60,000 CPM)
400	.150	.500		800 Hz (48,000 CPM)
350	.171	.500		800 Hz (48,000 CPM)
300	.200	.800		500 Hz (30,000 CPM)
240	.250	.800		500 Hz (30,000 CPM)
180	.333	1.000		400 Hz (24,000 CPM)
150	.400	1.000		400 Hz (24,000 CPM)
120	.500	1.333		300 Hz (18,000 CPM)
100	.600	2.000		200 Hz (12,000 CPM)
80	.750	2.667		150 Hz (9,000 CPM)
60	1.000	4.000	2048	200 Hz (12,000 CPM)
50	1.167	4.000		200 Hz (12,000 CPM)
40	1.500	4.000		200 Hz (12,000 CPM)
30	2.000	8.000		100 Hz (6,000 CPM)
20	3.000	8.000		100 Hz (6,000 CPM)
15	4.000	16.000		50 Hz (3,000 CPM)
12	5.000	16.000		50 Hz (3,000 CPM)
10	6.000	16.000		50 Hz (3,000 CPM)
6	10.000	32.000		25 Hz (1,500 CPM)
3	20.000	80.000		10 Hz (600 CPM)

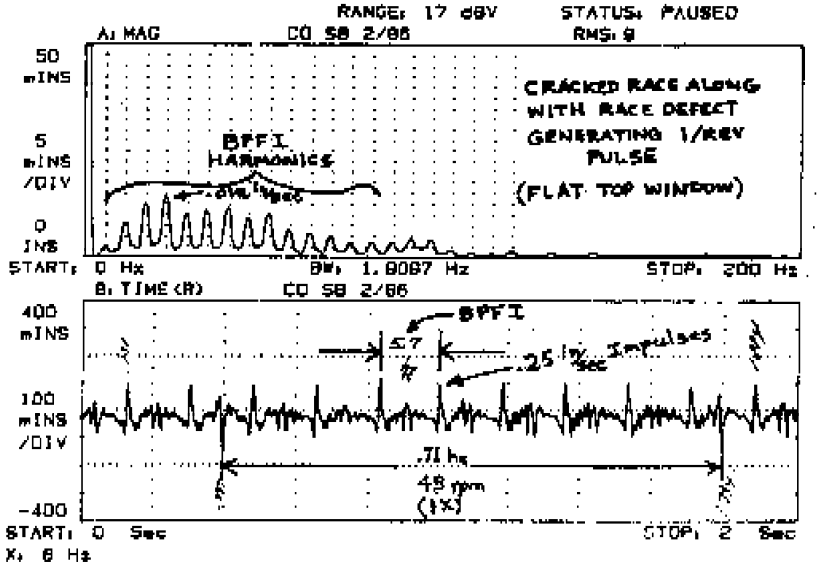

FIGURE 6.09J
CRACKED INNER RACE WAVEFORM & SPECTRUM FOR A 43 RPM ROLL
(Note the Pronounced Positive & Negative Pulses in the Time Waveform)
(Ref. 13)

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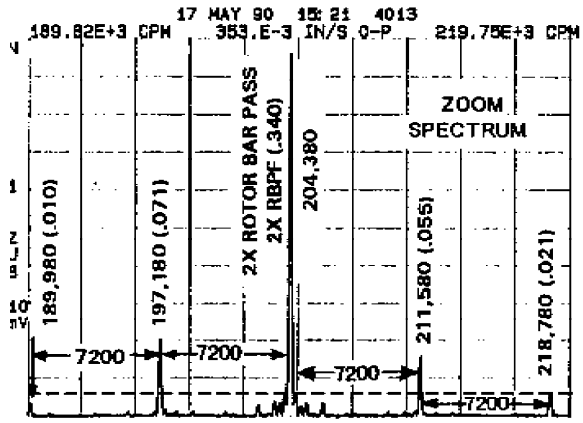

IN/S 0-P	CPM
70.9E-3	197.18E+3
21.0E-3	204.07E+3
340.E-3	204.38E+3
21.0E-3	204.60E+3
54.8E-3	211.58E+3

NOTE: 6-10-90 G.E. Repair Shop phoned to confirm this motor had 2 or more open rotor bars which will likely result in motor replacement

FIGURE 6.09K
DETECTION OF BEARING FAULTS USING THE TIME DOMAIN ON A TRUNNION BEARING OPERATING AT ABOUT 1.5 RPM (As Per Ref. 13)

13. Where Transducer Should be Placed to Properly Detect Bearing Frequencies:

It is most important to place the vibration transducer as close as possible to the load zone of the bearing, particularly if the bearing only supports radial load. Reference 13 points out that this is of critical importance for a spherical roller bearing in which he claims the vibration signal strength can be affected by as much as 100% if the transducer is not placed in the load zone, giving the analyst a false sense of security when he actually does have a problem.

14. Approximate Bearing Defect Frequencies if only the Number of Rolling Elements is Known:

Figure 6.09F provides 4 formulas which approximate the bearing defect frequencies if only the number of rolling elements (N_b) is known (as per Reference 18). These equations have been proven to be considerably more accurate than older approximations which assumed BPFO to equal approximately .4 x N_b x RPM and BPFI to equal approximately .6 x N_b x RPM. Still, optimum precision can only be realized if one knows each of the other bearing design parameters (B_d, P_d and Ø), particularly in the case of the ball spin frequency (BSF).

15. Sensing of Improper Bearing Load or Installation:

Not only can bearing frequencies be used to detect defects within the bearings, but also they can be used to detect when bearings are improperly loaded or installed. For example, even when a new bearing is installed, if there is excessive interference of the bearing seat on the bearing housing causing it to be “jammed into” the seat, it can result in take-up of all the internal clearances forcing the rolling elements against the races. If this occurs, the bearing will immediately generate ball pass frequencies of the outer and/or inner race upon startup. In addition, the author has taken data several times when thrust bearings were installed backwards. When this has occurred, the improperly installed thrust bearings have generated excessively high amplitudes at race defect frequencies, sometimes on the order of 1.0 in/sec, or greater. Excessive press fit of a bearing onto a shaft can also immediately generate a pronounced outer race or inner race defect frequency (BPFO or BPFI), letting the analyst

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know excessive and/or improper load has been placed on the bearing. And, even though no real wear might yet have occurred, if the problem is not detected and corrected, the bearing will likely fail long before its predicted life. In all improperly loaded situations, if the same bearings were reinstalled properly or loaded properly, the defect frequencies were significantly reduced or disappeared altogether. This has happened on a number of occasions. In fact, one client manufacturing textile machinery uses this technique to detect assembly problems in its quality assurance program.

16. Frequencies Generated by Inadequate Bearing Lubrication:

Reference 12 states that "unique signatures generated by inadequately lubricated bearings are characterized by 3 or 4 peaks in the frequency range of 900 to 1600 Hz (54,000 to 96,000 CPM). The difference frequency between the peaks ranges from 80 to 130 Hz (4800 to 7800 CPM). Some signatures of properly lubricated bearings contain these frequency components; however, the amplitude is very low - about 0.05 in/sec or less. The amplitude increases to as much as 0.10 or 0.20 in/sec when the lubrication is inadequate". He adds that "empirical evidence indicates that frequencies from 900 to 1600 Hz are natural frequencies of the installed bearing." With this in mind, if frequencies within this range (approximately 50,000 to 100,000 CPM) do occur with difference frequencies on the order of 5000 to 8000 CPM, it may be a good idea to check lubrication, particularly if high spike energy (or equivalent) levels are also measured on this same bearing housing. Also, even greater evidence of lubrication problems would occur if neither the high spike energy nor a spectrum having these components was present on the other bearing that is supporting this same shaft.

In addition, some research has shown that inadequately lubricated bearings which allow metal-to-metal contact can also generate bearing defect frequencies. In these cases, one might go ahead and add lubricant while monitoring both overall spike energy (or equivalent) and vibration FFT spectra to determine if the ultrasonic levels drop considerably, and to find if the bearing defect frequencies disappear from the spectrum. If they do, he should return to the machine 12 to 24 hours later and see if either have reoccurred. If not, lubrication was likely the problem. If they do return, the bearing is likely within one of the failure scenarios described in Section 6.093.

17. Defect Frequencies for So-Called "Equivalent Bearings":

The APPENDIX includes tables which allow one to determine "equivalent bearings" if he knows the AFBMA numbers of bearings within this machine (AFBMA numbers are often listed on the nameplates of many machines). As these tables show, this AFBMA number allows one to know several facts about the bearings including their bore, type (deep groove, angular contact, cylindrical roller, etc.), dimensional series (extra light, light, medium, etc.), type of cage or ball retainer, recommended lubricant and bearing tolerance/clearance. However, simply knowing the AFBMA number and corresponding "equivalent bearing" does not verify what bearing defect frequencies should be expected. Research has shown that so-called "equivalent bearings" have defect frequencies which can vary by as much as 22% due to differing numbers of rolling elements, different rolling element diameters, etc. That is, knowing that the AFBMA number is 65BC02JPP0 might mean that the bearing is either an SKF 6213, MRC 213S, Fafnir 213K, Norma-Hoffmann 213 or New Departure 3213 bearing. However, if one compared the actual bearing defect frequencies for these or other bearings, he will often find bearing frequencies which vary by a noticeable amount. Still, if only the AFBMA number is known, it is usually a good idea to at least record this number and to determine the equivalent bearings from the various manufacturers. Then, the analyst should list the bearing defect frequencies for 5 of 6 of these "equivalent bearings" so that he might have some idea where he might find the defect frequencies for these bearings, and how much the defect frequencies differ from one another.

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18. Electrically-Induced Fluting Problems Within Motor Bearings:

Within the last few years, it has been discovered that vibration signatures can be used to detect electrically-induced fluting within rolling element bearings. This has been found to be a widespread problem, particularly with the outboard (or opposite drive-end) bearings of DC motors. Reference 27 provides information from one author on how he has used vibration analysis to detect the presence of such electrical fluting and also relates he was surprised to find how little voltage can be present, and yet, do so much damage to the bearing surfaces. In fact, in one of his experiments, he found a voltage of only .4 to .5 volt passing through the bearing causing considerable damage. He also found the following when he had electrical fluting,

> "a spike that did not match any bearing frequency occurred between 40,000 - 60,000 CPM. Most often the spike was accompanied by the sidebands of whatever race had the biggest defect. It also could occur for both races. This was the one way we could routinely identify the electrical fluting problems. Another way to identify electrical fluting was listening to the motor itself. We noticed the motors sounded like they were under a heavy strain, yet the motors were under no load or under their normal load. As the bearing defect worsens, the sound intensifies and becomes deeper."

Figure 6.09L shows a velocity spectrum taken by this Reference 27 article taken on a DC motor bearing with electrically-induced fluting. His point here was that no real defect frequencies showed up at either BPFO or BPFI multiples. Instead, difference frequencies surrounded a significant spike which occurred at roughly 54,000 CPM (900 Hz). He reported the difference frequencies equalled the outer race and inner race frequencies in this spectrum.

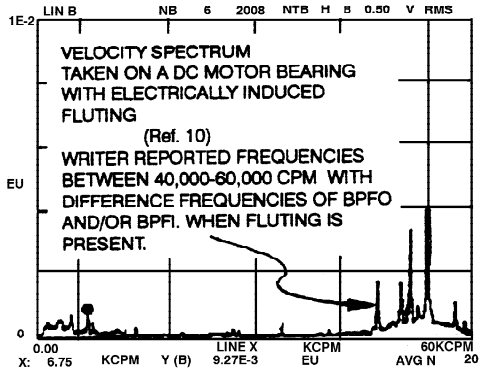

FIGURE 6.09L

Similar data was found by another analyst as shown in Figure 6.09M. Here, both velocity and spike energy spectra were taken on the outboard bearing of a DC motor having electrically induced fluting on its outer race. However, notice the great difference in the velocity and spike energy spectra. The velocity spectrum did not show any frequencies at the outer race frequency (BPFO), or BPFO harmonics. Instead, it showed a series of difference frequencies spaced at BPFO (approx. 4050 CPM) which were concentrated around peaks at 119,150 and 123,300 CPM. On the other hand, note the spike energy spectrum shown in Figure 6.09M-B which did show, in this case, 7 harmonics of outer race frequency

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(BPFO), with no other discernible frequencies in the spike energy spectra. One of the most important findings of these and other tests conducted by the author show that an analyst should capture at least one high frequency velocity measurement out to a frequency of approximately 180,000 CPM (3000 Hz) in order to detect possible electrically-induced fluting on DC motors. If he takes this data only in one location of a DC motor, it would likely best be taken on the outboard bearing. However, the author strongly recommends a similar high frequency measurement be taken on the inboard bearing (coupling side) as well (since higher frequency vibration does not often travel). Both measurements should be taken in the horizontal direction.

The writer of Reference 27 provided two solutions to the undesirable passage of electrical current through his motor bearings. In one case, the rear bearing on the motor was insulated by boring out the end bell of the motor and inserting a phenolic sleeve. Then the bearing was pressed back into the sleeve, thereby interrupting the path for current flow between the rear of the motor and the end bell. He reported this eliminated the electrical fluting problems in these bearings (however, the author has been told by several analysts that this solution works only about 80% to 90% of the time - not always). The Reference 27 analyst also tried installing carbon brushes on the end of the rear bearings where the shaft goes through the bearing. This resolved this particular problem, but he added that when the brushes wear, electrical current could once again pass through the bearings and do considerable damage if not detected soon enough.

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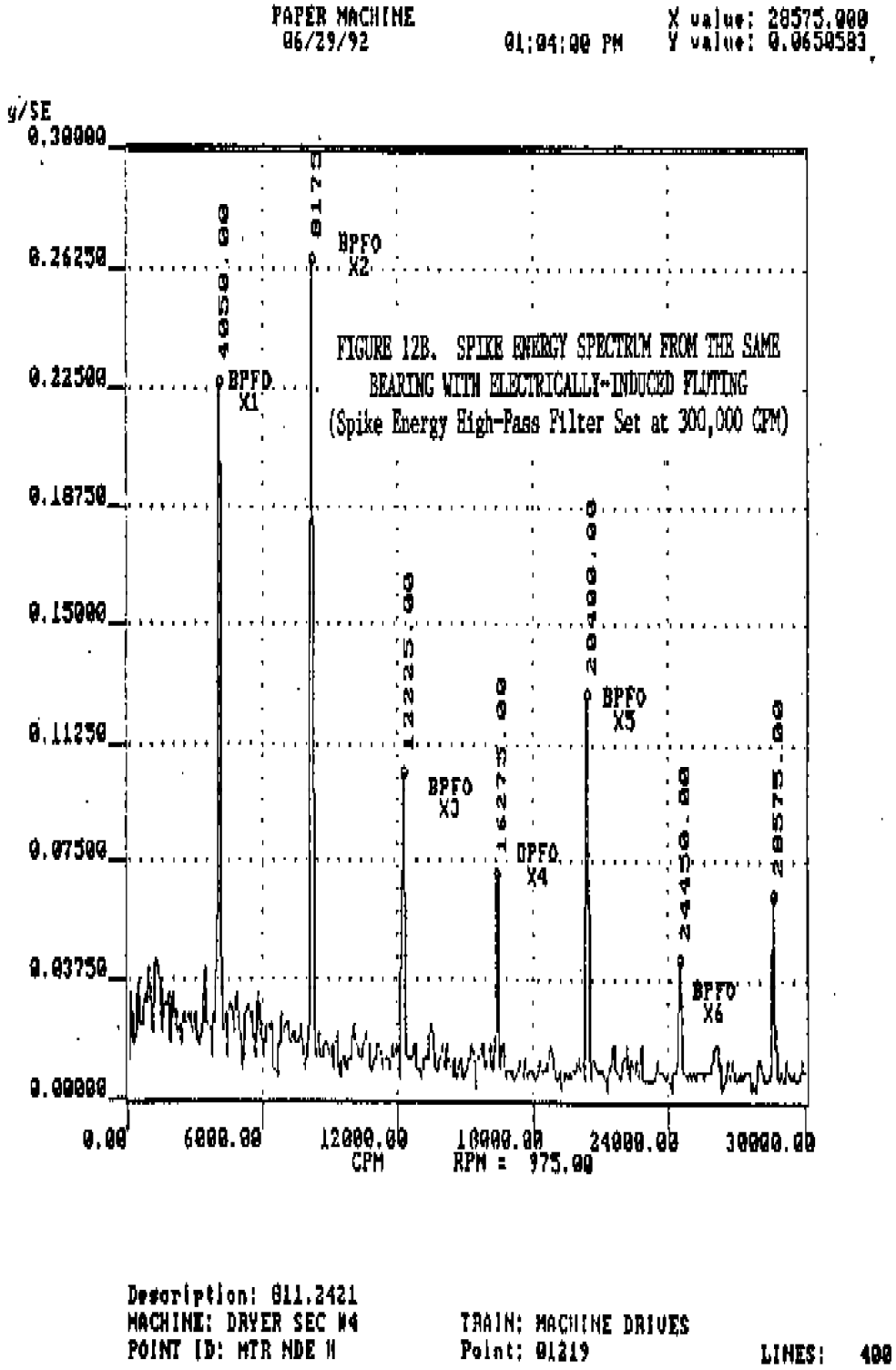

FIGURE 6.09M
COMPARISON OF VELOCITY & SPIKE ENERGY SPECTRA TAKEN ON THE OUTBOARD BEARING OF A DC MOTOR HAVING FLUTING ON ITS OUTER RACE DUE TO PASSAGE OF ELECTRICAL CURRENT THROUGH THE BEARING

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(d) Sum and Difference Frequencies:

When a single defect has developed on a bearing, it will generate a defect frequency and will begin deteriorating. When the defect grows, it can contribute to the development of other defects in the bearing. In so doing, other frequencies will be generated and a number of patterns might develop. Some frequencies will add to and subtract from others. In fact, the fundamental frequency for a particular defect might never occur. When it appears, this defect frequency might act as a sideband to other frequencies which are already present. For example, a cage frequency itself will not normally occur at its fundamental frequency of approximately .35X to .45X RPM. Instead, the cage frequency itself normally will sideband the race frequencies (BPFO or BPFI) or the ball spin frequency of the bearing (see Figure 6.09I). In some cases, bearing defect frequencies can even modulate frequencies generated by sources other than bearings. Following below are some of the more important facts concerning sum and difference frequencies and their behavior:

1. It is not uncommon for the fundamental ball spin frequency or harmonics never to appear even if significant faults are present on the rolling elements. In this case, BSF will appear, but as a sideband of other frequencies.

2. If only a single fault is present on either the outer or inner race, only a single race frequency will appear. However, when faults begin to appear around the periphery, a number of harmonics of these race frequencies will appear.

3. When defects grow on raceways, the amplitude of the race frequencies themselves will often increase somewhat. However, even greater indicators of deterioration are the number of bearing frequency harmonics, as well as the appearance of 1X RPM sideband frequencies above and below the race frequencies (particularly BPFO). In effect, the unbalance forces at shaft speed will tend to modulate the frequency components of the races, generating sidebands. In the case of BPFI, 1X RPM sidebands are often created around it since its amplitude is modulated with much greater response when the inner race defect impacts rolling elements within the load zone than those generated outside the load zone. The rate of this modulation in amplitude is at 1X RPM, therefore generating the 1X RPM sidebands (see Figure 6.09H).

4. In rolling element bearing frequency analysis, the emphasis is <u>not on amplitude</u>, but on <u>content of the spectra</u>. In fact, <u>amplitudes of bearing frequencies themselves often begin to drop as condition worsens</u>, particularly in the case where faults begin to propagate around the periphery of the outer or inner race. In this case, serious weight should be placed on the fact that a multiple number of fault frequencies are appearing and that many of these fault frequencies are sidebanded by vibration at bearing RPM.

5. As deterioration continues, modulation effects can continue to have more and more influence until eventually, the spectrum can become a series of nothing but 1X RPM harmonics.

6. Reference 14 shows that double row bearings having staggered rolling elements can generate 2 sets of bearing defect frequencies - (a) If a defect is confined to only one side of the raceway, the bearing defect frequencies generated should be calculated using the number of rolling elements only in that single row; however, (b) if a defect occurs on both sides of the raceway, the frequencies should be calculated using the total number of rolling elements. Here again, either of these frequencies can act as a sideband around other frequencies with the sideband difference depending on whether the fault has occurred on one or both raceways. Importantly, this number of balls (either those for one row or the total number) must be entered into the outer race, inner race and ball spin frequencies.

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7. A single defect on the inner race of a rolling element can be difficult to diagnose from frequency spectra alone since amplitudes will most often be low, and discrete spectral lines might not be seen at calculated defect frequencies. This particularly applies to bearings with inner race rotation mounted with an interference fit between the inner race and the shaft. In these cases, the load zone might be noticeably less than 180° which explains why discrete peaks at defect frequencies may not appear. Reference 15 points out the real problem occurs if a crack appears on the inner race. In these cases, the shaft fit can be relieved and internal clearances of the bearing lost. He points out that such a situation is extremely dangerous which can bring about rapid seizure of the bearing causing the inner race to spin on the shaft or the outer race to spin in the housing resulting in catastrophic failure. In these cases, since the inner race frequency itself or harmonics may not show, but act as a sideband on other frequencies, he recommends referring to the time domain (or time waveform) in addition to the vibration spectra themselves.

8. In rolling element bearings, when multiple running speed harmonics are present, they can signal either looseness of the bearing on the shaft or within its seat or, much more importantly, they can likewise signal a bearing turning on the shaft or in the housing. The author has been involved in several situations where a number of very low amplitude running speed harmonics (less than .04 in/sec) were present which resulted in catastrophic failures, even with only the low amplitudes. In many of these cases, there were in fact no real defects within the bearings themselves, but the bearing turning on the shaft resulted in severe shaft damage (in one case, over .25 inch diameter was lost before eventual failure).

Thus, there is great information present within the signatures of rolling element bearings that will help identify their current condition and assist the analyst in recommending possible corrective actions. Following in Section 6.093 will be a presentation of a series of typical spectra which can be used by the analyst to track the condition of rolling element bearings under some of the most common failure scenarios which have occurred in the experience of the author.

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6.093 TYPICAL SPECTRA FOR TRACKING FAILURE STAGES THROUGH WHICH ROLLING ELEMENT BEARINGS PASS:

During the last decade, concentrated research and experimental investigations have been conducted on how to best evaluate rolling element bearing condition using vibration analysis and high frequency enveloping techniques. Some of the better papers which have been written include References 2, 12 and 19. In addition, Technical Associates has been deeply involved in further developing rolling element bearing diagnostic capabilities. We have consulted with a variety of clientele on a wide array of machinery ranging from massive, low-speed machines outfitted with large, expensive rolling element bearings to very small, high speed rolling element bearing machinery.

Experiments have been conducted purposely running rolling element bearings to failure while carefully tracking them with vibration signatures (Reference 11). At various intervals, the bearings were broken open and closely inspected for the extent of damage, the type of deterioration and the probable cause for the onset of faults within the bearings. These results would then be closely correlated with the vibration spectra, along with various ultrasonic frequency measurements (i.e., Spike Energy, Shock Pulse and HFD). The bearings were purposely opened up and examined when such data indicated them to be in a wide range of condition (from those with supposedly no faults to those indicated to have extensive damage). Some only had increases in ultrasonic measurements (not in vibration); others showed increases only in high frequency regions; while clearly defined bearing defect frequencies were within vibration spectra of others (some were allowed to catastrophically fail while vibration and ultrasonic responses were closely tracked). Such experiments were performed on a variety of rolling element bearings at different speeds and conditions. This work has enabled the development of fairly well proven diagnostic methods for the tracking of rolling element bearing condition using vibration signature analysis, along with time waveform and ultrasonic analysis.

It was found that the majority of rolling element bearings followed a fairly predictable failure path from the very onset of deterioration through eventual catastrophic failure. This failure path is graphically portrayed in Figure 6.09N which plots bearing damage versus time. Note importantly that bearing damage typically will increase exponentially during the final 10% to 20% of its life. It is here where intensive research and field investigations have found how to use vibration analysis and high frequency enveloping tools to identify “failure stages”. From this, a classic 4 stage failure scenario has been developed which will apply to approximately 80% of rolling element bearing failures. This will be presented in Figure 6.09P as Scenario A entitled “4 Primary Failure Stages Through Which Most Rolling Element Bearings Pass”. Text describing each of the 4 failure stages will accompany Figure 6.09P.

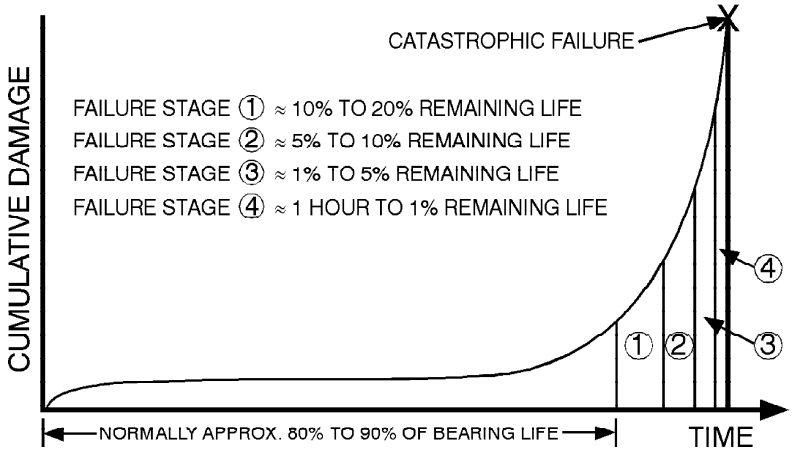
CATASTROPHIC FAILURE
FAILURE STAGE ① ≈ 10% TO 20% REMAINING LIFE
FAILURE STAGE ② ≈ 5% TO 10% REMAINING LIFE
FAILURE STAGE ③ ≈ 1% TO 5% REMAINING LIFE
FAILURE STAGE ④ ≈ 1 HOUR TO 1% REMAINING LIFE
CUMULATIVE DAMAGE
①
②
③
④
NORMALLY APPROX. 80% TO 90% OF BEARING LIFE
TIME

FIGURE 6.09N
TYPICAL FAILURE PATH TAKEN BY ROLLING ELEMENT BEARINGS

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As was expected, not all rolling element bearings followed the classic failure path (Figures 6.09N and 6.09P). Therefore, this paper will attempt to both illustrate and document the other failure paths which were taken by a much smaller percentage of rolling element bearings which will likewise be illustrated and documented in Scenarios B through F, respectively.

It is difficult to categorically state the exact percentage of total failures represented by each of the six failure scenarios presented. However, at this time, it appears that Scenario A alone probably represents approximately 80% of all rolling element failure paths. Therefore, that for all the remaining five failure scenarios represents only approximately 20% of the total population.

SCENARIO A. 4 PRIMARY FAILURE STAGES THROUGH WHICH MOST ROLLING ELEMENT BEARINGS PASS (FIGURE 6.09P):

Figure 6.09P presents typical velocity spectra for each of the 4 failure stages for most rolling element bearings. These spectra follow the bearing from the very onset of bearing problems in Stage 1 through imminent failure of the bearing in Stage 4 (see Figure 6.09N). Note that the overall spike energy (or HFD) amplitudes given to the right of each spectrum are meant to be rough approximations only. Also, note that documentation below the title of each one of the stages refers to an approximate "L_{10} Life Remaining" which corresponds to the approximate remaining anticipated life of the bearing based on a 90% confidence level (L_{10} Life is documented and its formula is given in the introductory discussion on page 1). Today, in general, the machine designer normally will attempt to provide a design life of approximately 5 to 10 years for most common machinery. Therefore, when a percentage on the order of 10% of L_{10} Life is quoted below, this will generally mean from 6 months to 1 year remaining life anticipated (depending on the type of machine and its intended application). Of course, this can vary widely with the machine type, with particular design parameters and whether or not the bearings are provided with the proper lubrication, operating temperature and subjected to acceptable vibration levels.

It is important to point out that bearings do not follow a linear deterioration path, but instead tend to fail exponentially; that is, when the bearing for example enters Stage 3 outlined below, it may still have a fair remaining life, or it may fail rapidly. Once the bearing enters Stage 3, the failure can progress rapidly, particularly towards the end of this stage. Generally, low speed bearings may still have fairly predictable remaining life (unless exposed to high dynamic loads - see L_{10} Life Formula discussion on page 1 which shows that bearing life is inversely proportional to the 3rd power of imposed loading).

Table 6.09A compiled by Charles Berggren in Reference 16 (reprinted here with his permission) roughly adheres to the 4 stages illustrated in Figure 6.09P and documented below:

Stage 1 (Approximately 10% to 20% L_{10} Life Remaining):

Spike energy (or equivalent) normally first appears before any frequencies do so in the vibration spectrum (velocity or acceleration). For example, Stage 1 shows a normal spectrum indicating a healthy bearing and has only the normal first 3 running speed harmonics in the velocity spectrum. The only evidence of possible bearing problems is that Spike Energy has grown from near 0 to approximately .25 gSE (example amplitude only; actual levels depend on the particular bearing and how close the measurement is to the bearing housing). Note that the Spike Energy reading itself is not meant to be part of the spectrum in Figure 6.09P, but only an overall level. During Stage 1, no sound will be detectable by the human ear indicating bearing damage, and no change in bearing temperature would be anticipated. Table 6.09A shows the defects that would be expected in this stage, many of which would be near microscopic at this stage.

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FIGURE 6.09P
4 PRIMARY FAILURE STAGES THRU WHICH MOST ROLLING ELEMENT BEARINGS PASS (VELOCITY SPECTRA) (Scenario A)

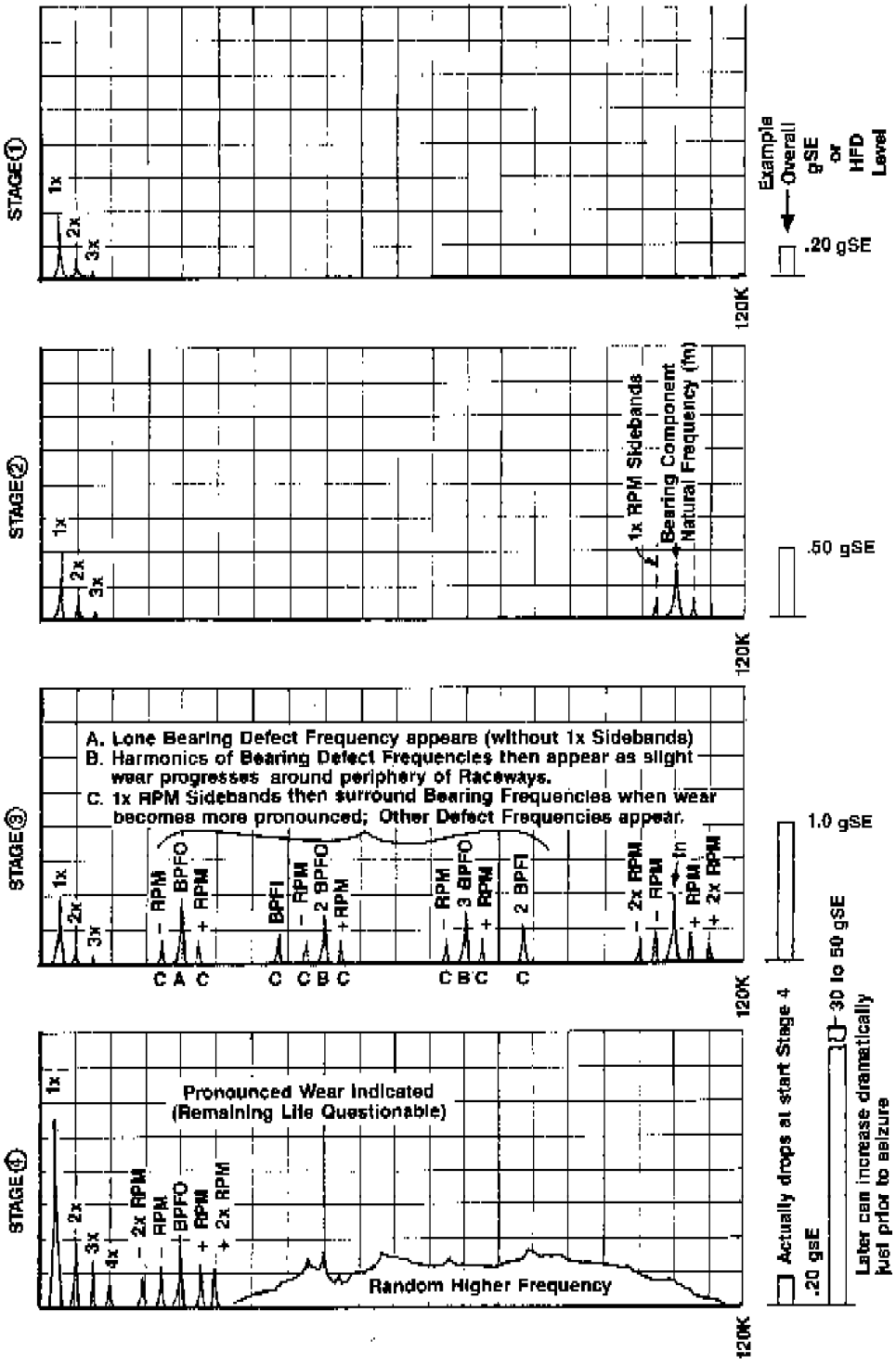

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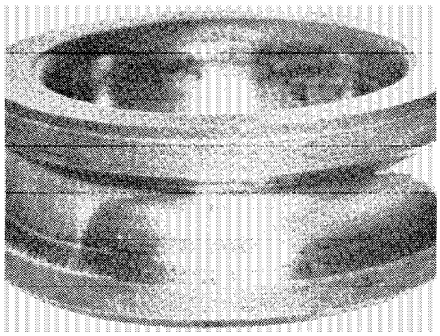
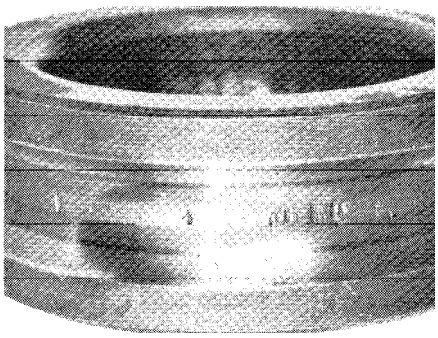
TABLE 6.09A
4 ROLLING ELEMENT FAILURE STAGES (Reference 16)
Typical Stages Of Bearing Degradation

a. Initial Phase.



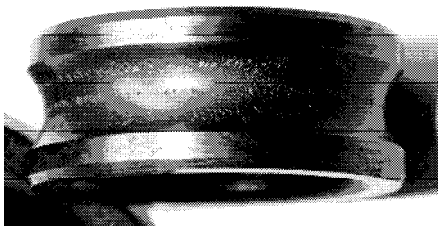
- noise level normal
- temperature normal
- measurable increase in ultrasonic sound, acoustic emission, spike energy, and outer race deflection
- overall vibration low; no discrete spikes at bearing
- frequencies remaining life more than 10% of B-lO rating*

b. Second Phase.



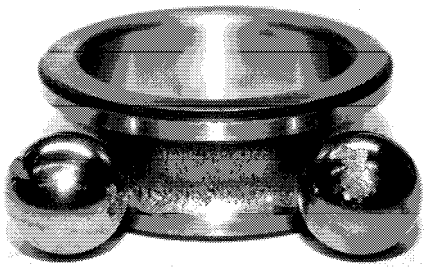
- slight increase in noise level
- temperature normal
- large increase in ultrasonic sound, acoustic emission, spike energy, and outer race deflection
- slight increase in overall vibration acceleration and velocity
- bearing frequencies clearly visible on log scale, barely visible on linear scale of vibration spectrum; noticeable rise in noise floor
- remaining life less than 5% B-10 rating

c. Third Phase.



- noise level quite audible
- slight increase in temperature
- very high ultrasonic sound, acoustic emission, spike energy, and outer race defection
- large increase in overall vibration acceleration and velocity
- bearing frequencies with harmonics and sidebands clearly visible on linear scale
- of vibration spectrum; noticeable rise in noise floor
- remaining life less than 1% B-10 rating*

d. Final Phase.



- change in pitch of noise level
- significant temperature increase
- gradual decline followed by rapid increase in ultrasonic sound, acoustic emission, spike energy, and outer race deflection immediately prior to failure
- significant increase in overall vibration displacement and velocity; decrease in acceleration
- vibration spikes predominant at lower bearing frequencies; very high noise floor in spectrum remaining life less than 0.2% B-10 rating*

*based on 90% confidence level

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Figure 6.09Q shows what would be referred to as a classic "Failure Stage 1" scenario. This data was taken on a 1790 RPM combustion blower at a fiberglass plant where it was one of the most critical machines. First, note the trend of overall spike energy which increased from 1.43 up to 7.03 gSE on this inboard blower bearing (note that this data was taken with an IRD 943 accelerometer connected to an IRD FAST TRACK/FS analyzer; therefore, the spike energy values tend to be higher with the model 943 than they formerly did with the older IRD 970 accelerometer; in any case, the significant spike energy trend signaled a problem). Examination of the spectral history and the latest spectrum did not reveal any frequencies associated with the bearings within this machine. Therefore, referring to failure Scenario A in Figure 6.09P, this would place this bearing within Stage 1 if the increase in spike energy was due to bearing damage (since increased spike energy levels can not only be due to bearing problems, but also to lubrication, gear problems, rub, cavitation and many other "friction-related" events which can generate very high frequencies).

It is exactly at this point that vibration analysts were stymied for a number of years. Ever since the development of Spike Energy (and related parameters such as Shock Pulse and Acoustic Emission), analysts had wondered if the bearing truly had a problem, why was nothing apparently showing up in either vibration velocity or acceleration spectra. And another question was asked after microscopic evaluations of the bearing component surfaces were conducted after measurements showed an increase in one of these ultrasonic quantities (when it was confirmed the increase was indeed due to bearing problems). Finally, only a few years ago, metallurgists found the answer. Even though the component surfaces themselves showed little or no apparent damage, the metallurgists discovered an important fact - if the bearing is failing properly in fatigue, the damage most often will begin approximately 4 to 6 mils (.004 - .006 inch) beneath the surface and will work its way out to the surface. Initially, there will only be damage to the actual crystalline structure of the bearing materials themselves beneath the surface. Later, a microcrack will work its way to the surface, but will still be microscopic in size. However, the important fact is that the damage mechanism has begun, and cumulative damage from this point onwards will likely be exponential. When the rolling elements roll past and impact this microscopic flaw, they will generate a very short-lived transient impulse, normally generating a response less than only .001 g. Therefore, the simple fact is that vibration due to mechanical and electrical deviants such as unbalance, misalignment, blade pass, rotor bar problems, etc., will easily mask (or cover up) these responses in the lower frequency ranges (particularly below 120,000 CPM or 2000 Hz), making it impossible to see the actual bearing defect frequencies which indeed are there, but are "invisible" to the vibration instruments and spectra.

However, it is well known that very short-lived transients generate numerous frequency harmonics (resembling "white noise" causing the harmonics to virtually "propagate forever", but still of very small amplitude. However, since they do propagate all the way into very high frequencies on the order of 20,000 to 40,000 Hz (2,400,000 - 4,800,000 CPM), they eventually will excite the resonance either of the transducer itself measuring the ultrasonic response, or of the surrounding structure or some other member. It is within these frequencies that such parameters as Spike Energy, Shock Pulse and Acoustic Emission do their detection work. As a matter of fact, each of these parameters purposely employs high pass filters to essentially remove the contribution in the spectrum from the mechanical and electrical sources mentioned above which were masking the bearing problem.

Some of the new data collection systems now employ high frequency enveloping and demodulation techniques which can now work with the remaining data in the high frequency ranges. In essence, they use the time waveform (which might include the numerous bearing frequency harmonics as well as the natural frequency of one of the transducer or structural components); full-wave rectify the modulated waveform; determine the spacing of the exciting frequencies (bearing frequencies in this case); and then pass the remaining waveform data through a low-pass filter to remove the high frequency resonant response or so-called "carrier

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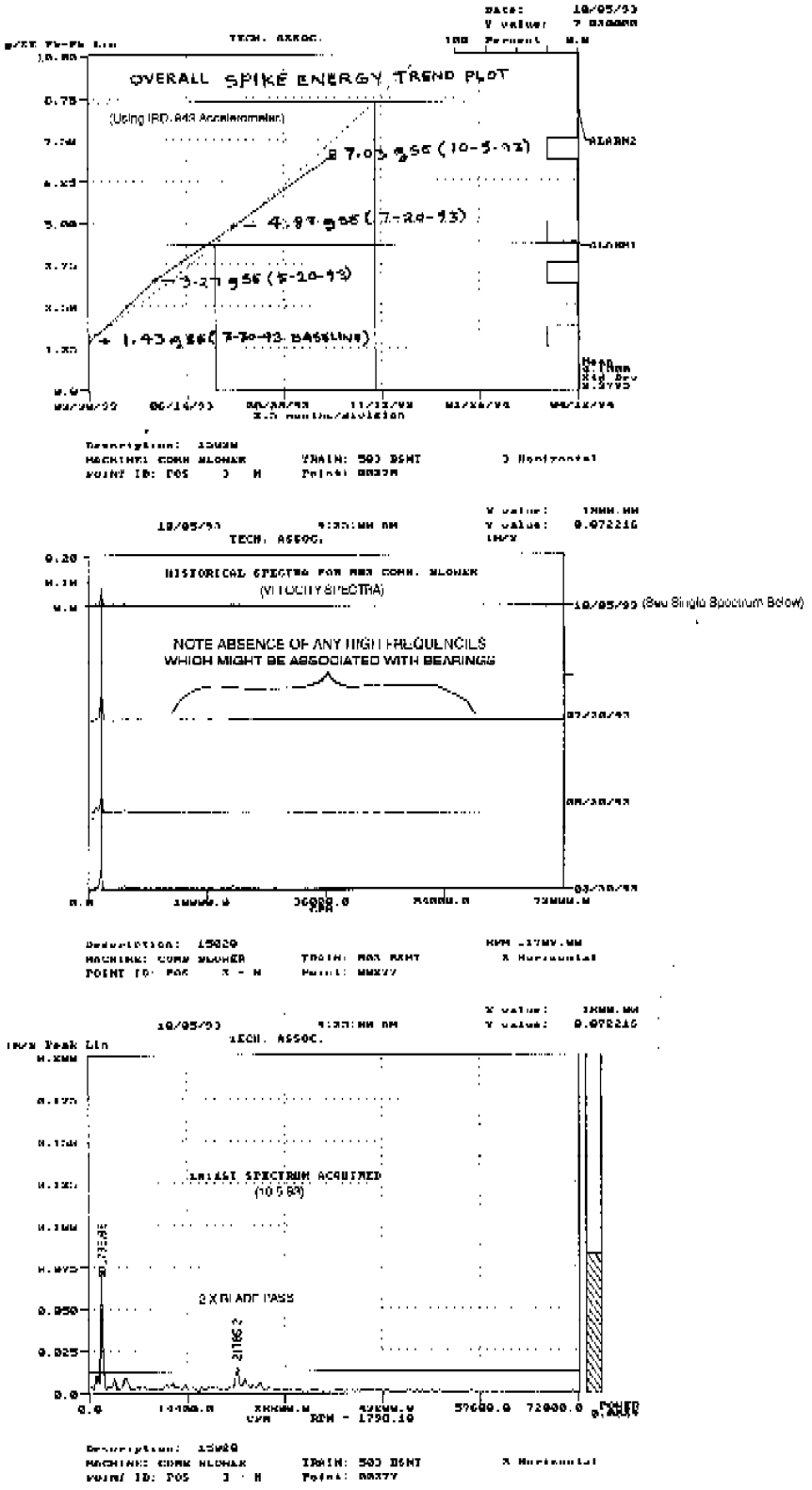
FIGURE 6.09Q
CLASSIC STAGE 1 APPEARANCE OF SPIKE ENERGY TREND AND ACCOMPANYING VELOCITY SPECTRA



frequency". The purpose here will be to see what frequencies were exciting the resonance and, using the low-pass filter, to fold them back into a lower frequency span to identify these frequencies. If in fact they were bearing frequencies, it would point to the bearing as a problem. Now, the problem might not necessarily be wear at this time. Instead, it might only be a lack of lubricant for example. Therefore, at this point, the analyst should add lubricant to the bearing and repeat these ultrasonic spectra as well as measurements of the overall levels. If they drop considerably or even disappear, he should then repeat the measurements within 12 to 24 hours to see if they return. If they remain of little or no amplitude, the problem was likely lubrication and the bearing has now been given a chance to hopefully enjoy a significantly increased life had the lubrication problem not been caught. On the other hand, if the bearing defect frequencies have reoccurred, we are likely within Stage 1.

Figures 6.09R through 6.09T give examples of such so-called ultrasonic spectra with data acquired from each of three vendors on various machines. Figure 6.09R illustrates detection of a Stage 1 bearing problem by spike energy spectra which were not present within the vibration spectrum alone (this data was taken with an IRD FAST TRACK/FS analyzer). Looking at the velocity spectrum in Figure 6.09R-A, note where outer race frequencies (BPFO) would be present if they were in the velocity spectrum. You will note that although cursors have been placed in these locations, there seems to be no real response due to the masking problem previously discussed. However, examination of the spike energy spectrum in Figure 6.09R-B clearly showed the presence of 5 BPFO harmonics on this 3590 RPM vacuum blower which would have totally been missed had vibration spectra alone been taken. When the same frequencies remained after lubrication, this confirmed the onset of failure Stage 1 for this bearing.

Similarly, Figure 6.09S used another analyzer (SKF CMVA10 analyzer) to detect bearing problems not visible within the vibration spectrum. In this case, it used not only what are known as acceleration envelope techniques, but also the new SEE spectral technology to confirm the bearing problems ("SEE" stands for "Spectral Emitted Energy" and was developed by SKF). This data was taken on a 1250 RPM belt-driven fan bearing. Figure 6.09S-A is the acceleration spectrum on which are overlaid a series of 8 outer race frequency markers which showed where such outer race frequencies would be if they were in the acceleration spectrum (which they were not). On the other hand, Figure 6.09S-B used a bandpass filter of about 5000 to 40,000 Hz and employed acceleration enveloping with the SKF CMVA10 analyzer and did show that both outer race and inner race frequencies were located after using the bandpass filter of 5,000 to 40,000 Hz (300,000 - 240,000 CPM) to filter out the lower frequency mechanical and electrical problems. Similarly, Figure 6.09S-C employed SKF's new SEE technology to confirm the bearing problems seen in the acceleration envelope. Currently, the SEE spectrum uses a much higher bandpass filter of approximately 250,000 to 350,000 Hz (15,000,000 - 21,000,000 CPM). In this case, note that the bearing problems were even clearer in the SEE spectrum with higher signal to noise ratio than was the case with the acceleration envelope. Note that further discussion of the SEE spectrum and these other related high frequency enveloping and modulation technologies is provided in Reference 28.

Figure 6.09T demonstrated the fact that even these high frequency demodulation technologies can be employed on very low speed machines. In fact, they will often detect problems which cannot even be detected by either the FFT spectrum nor the time waveform when using vibration measurements alone. In this case, the measurements shown in Figure 6.09T were taken on a 22 RPM (.366 Hz) dryer roll which had a bearing frequency of 3.125 Hz (187.5 CPM). In this case, a CSI 2115 analyzer was outfitted with a Model 750 Amplitude Demodulator which was attached to the 2115 providing it with this amplitude demodulation capability. First, vibration data shown in Figures 6.09T-A and 6.09T-B did not show any evidence of a bearing problem. Figure 6.09T-A was the time waveform with a time span of 4.0 sec (corresponding to almost 1.5 roll revolutions). It showed the presence not only of the running speed, but evidence of higher frequencies as well. However, it would have been extremely difficult to pick out the inner race frequency from this time waveform. Figure 6.09T-B showed the FFT spectrum corresponding to the time waveform

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FIGURE 6.09R
DETECTION OF STAGE 1 BEARING PROBLEM BY A SPIKE ENERGY SPECTRUM WHICH WAS NOT INDICATED BY THE VIBRATION SPECTRUM ALONE (Used IRD FAST TRACK/FS Analyzer)

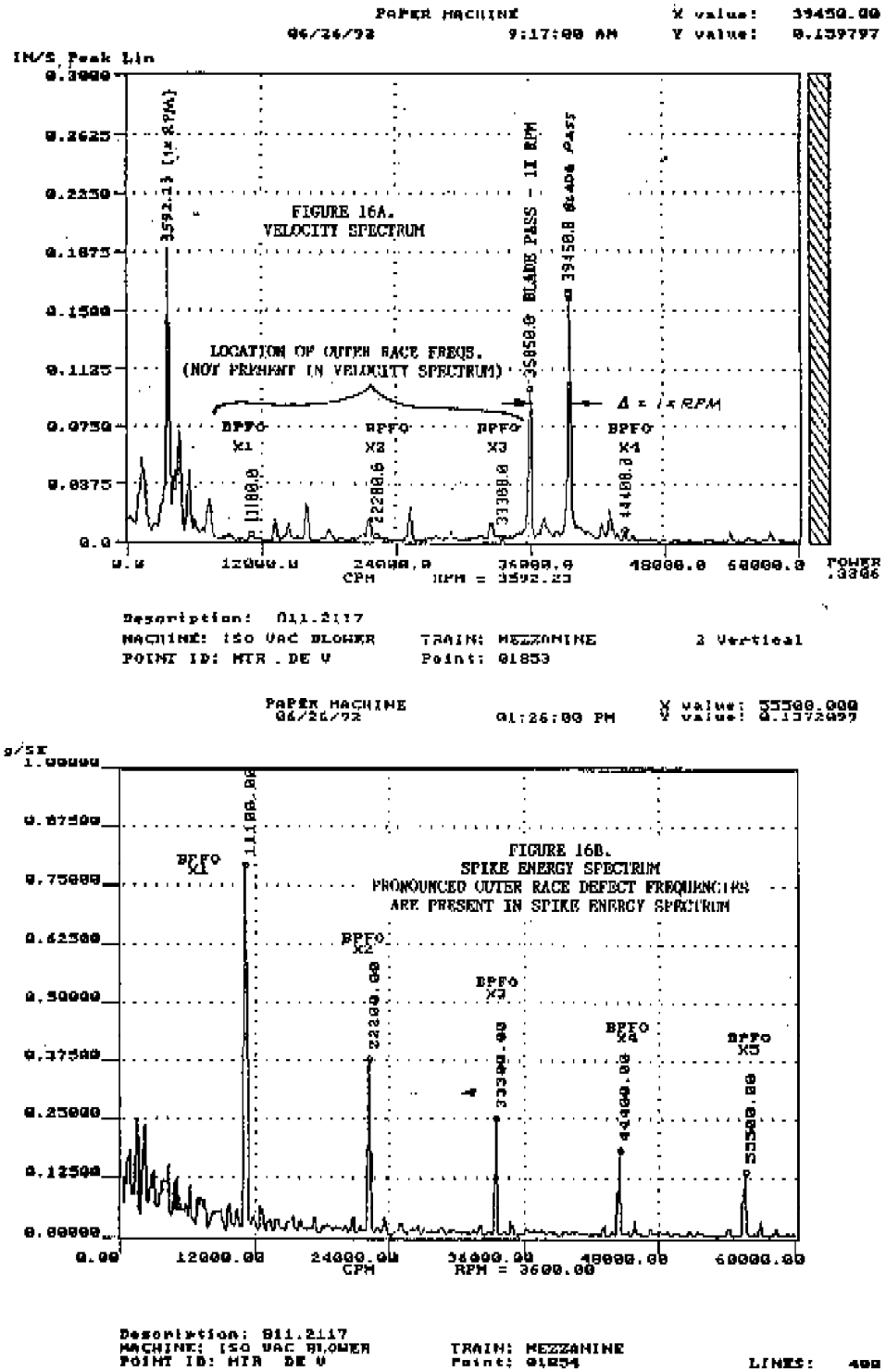

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FIGURE 6.09S
DETECTION BEARING PROBLEMS BY ACCELERATION ENVELOPE AND SEE SPECTRAL TECHNIQUES WHICH WERE NOT EVIDENT WHATSOEVER IN THE VIBRATION SPECTRUM (Used an SKF CMVA10 Analyzer)

SET: MULTILOG 1 TYPE: FFT DATE: 12-NOV-92 16:30:58
POINT ID: POINT 1 ACC LABEL 2 DESC: Acceleration on Bearing A
WINDOW: HANNING LINES: 400 AVER: 1 FREQ: 0 - 1000 Hz
DETECT: RMS RPM: 1250 THRESHOLD: 0.1000 UNITS: Gs

FREQ: 20.00 AMP: 0.004 ORDER: 0.960 DEG: ---

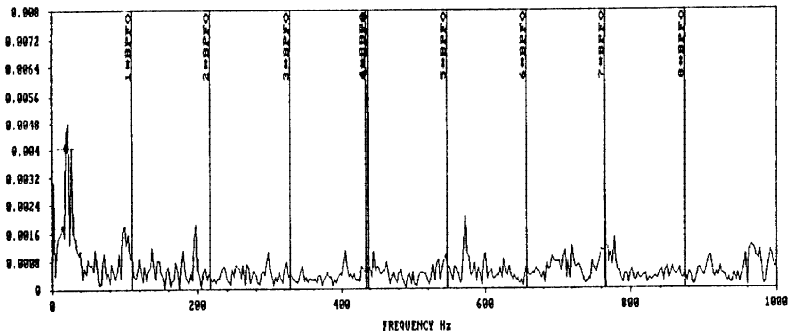

FIGURE A. ACCELERATION SPECTRUM TAKEN ON A BELT-DRIVEN FAN BEARING (No Evidence of Bearing Problems)

SET: MULTILOG 1 TYPE: FFT DATE: 12-NOV-92 16:33:09
POINT ID: POINT 1 ENV LABEL 2 DESC: Envel Acceleration on Bearing
WINDOW: HANNING LINES: 400 AVER: 8 FREQ: 0 - 1000 Hz
DETECT: PEAK TO PEAK RPM: 1250 THRESHOLD: 0.4000 UNITS: Gs

FREQ: 17.50 AMP: 0.2565 ORDER: 0.840 DEG: ---

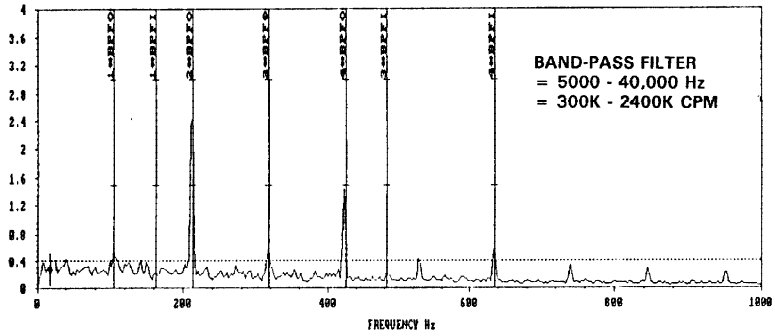

FIGURE B. ACCELERATION ENVELOPED SPECTRUM SHOWING OUTER RACE (BPFO) AND INNER RACE (BPFI) PROBLEMS

SET: MULTILOG 1 TYPE: FFT DATE: 12-NOV-92 16:32:44
POINT ID: POINT 1 SEE LABEL 1 DESC: See Meas on Bearing A
WINDOW: HANNING LINES: 400 AVER: 8 FREQ: 0 - 2000 Hz
DETECT: PEAK TO PEAK RPM: 1263 THRESHOLD: 6.0000 UNITS: SEEs

FREQ: 20.00 AMP: 10.87 ORDER: 0.950 DEG: ---

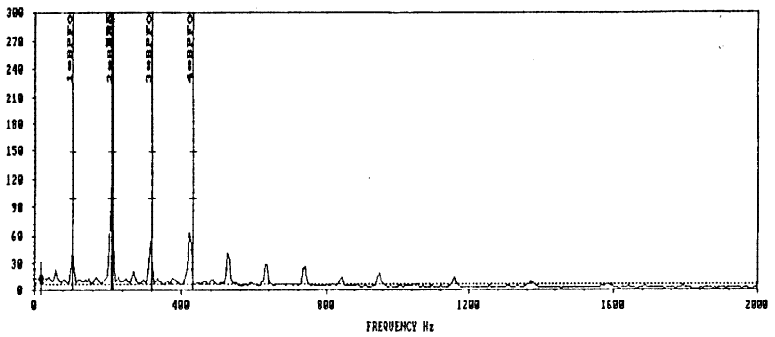

FIGURE C. SKF'S SEE SPECTRUM CONFIRMING THE BEARING PROBLEMS SEEN IN THE ACCELERATION ENVELOPED SPECTRUM

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FIGURE 6.09T
DETECTION OF INNER RACE PROBLEMS ON A 22 RPM DRYER ROLL BEARING BY AMPLITUDE DEMODULATED SPECTRA WHICH WERE COMPLETELY MASKED BY BOTH VIBRATION TIME WAVEFORM AND FFT SPECTRA (Ref. 29)
(Used a CSI 2115 Analyzer and CSI Model 750 Amplitude Demodulator)

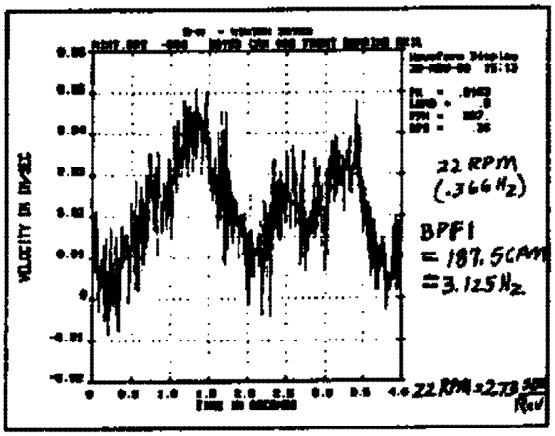

FIGURE A
RAW VIBRATION VELOCITY WAVEFORM FROM A DRYER BEARING

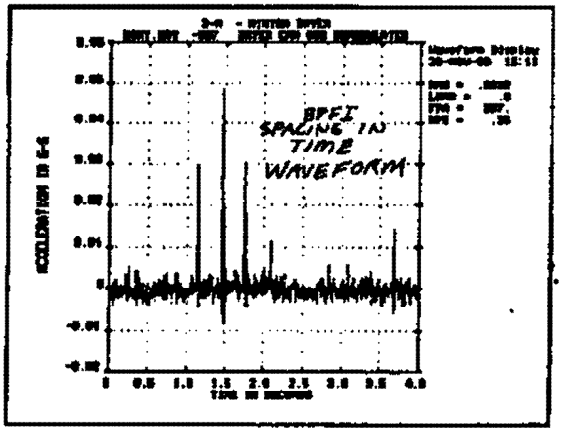

FIGURE C
DEMODULATED VIBRATION ACCELERATION WAVEFORM FROM WHICH THE SPECTRUM OF FIGURE D IS DERIVED

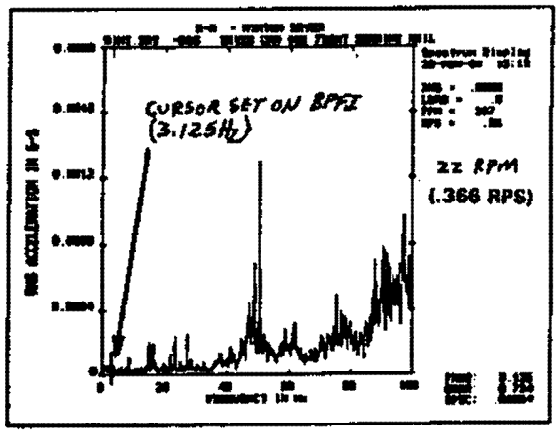

FIGURE B
VIBRATION ACCELERATION SPECTRUM DERIVED FROM WAVEFORM IN FIGURE A

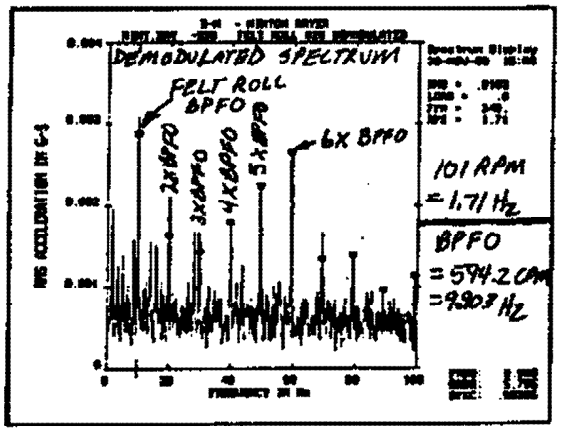

FIGURE D
DEMODULATED VIBRATION ACCELERATION SPECTRUM OF THE SAME BEARING SHOWN IN FIGURES A AND B

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captured in Figure 6.09T-A. In this case, the cursor was purposely positioned on the inner race frequency (BPFI) of 3.125 Hz and showed almost no response. The only major peak showing within the spectrum was at approximately 60 Hz which corresponded to the gear mesh frequency, and even it had an amplitude of only .0012 g. Figures 6.09T-C and 6.09T-D showed the resulting spectral and waveform data which resulted after the Model 750 Amplitude Demodulator was invoked. First, both 1X and 2X BPFI were evident in the FFT spectrum (note that the amplitude at BPFI was only .0005 g). In addition, note the clear BPFI spacing in the amplitude demodulated time waveform in Figure 6.09T-D. It has been the experience of the author that in many cases, neither the FFT spectrum nor the time waveform showed any evidence of bearing problems. However, employing high frequency spectral techniques such as this amplitude demodulated spectrum and waveform provided just the evidence needed to prove there was indeed a problem. In a few rare cases, particularly on machines running below 20 RPM, nothing ever did show up in either the vibration waveform nor spectrum, but did show in the high frequency enveloped and demodulated spectra, thereby preventing the failure of bearings costing tens of thousands of dollars. Further information is provided on high frequency enveloping and demodulation spectral techniques in Reference 11. The important point to be made here is that these technologies will not replace vibration, but serve as invaluable tools to confirm the presence of bearing problems; and in some cases, serve as the only detection tools available on very low speed machines with bearing problems.

Stage 2 (Approximately 5% to 10% L_{10} Life Remaining):

Slight bearing defects begin to "ring" natural frequencies of the installed bearing components. These natural frequencies are concentrated in the 30,000 to 120,000 CPM range (500 to 2000 Hz). Fundamental natural frequencies of most bearings in industrial machinery are approximately 30,000 to 90,000 CPM (500 - 1500 Hz) while those for high precision bearings normally range between 75,000 to 120,000 CPM (1250 - 2000 Hz); and on such high precision machine tooling, may occasionally range as high as 150,000 to 180,000 CPM (2500 - 3000 Hz). Remember, this is not one individual natural frequency, but a set of natural frequencies for the bearing outer race/ inner race/rolling element/housing system. These are natural frequencies of the assembled rolling element bearings themselves which do not change in frequency with a change in operating speed (however, they normally will show higher amplitudes with increasing speed due to greater impact velocity). These natural frequencies are excited by the momentary impact between the rolling elements and bearing races which not only excite the bearing natural frequencies, but also increases ultrasonic response (for example, roughly doubling the overall ultrasonic level in many cases). It has been the experience of the author that during initial Stage 2, only one or more discrete frequencies appear in these regions. Later, towards the end of Stage 2, these frequencies will not only grow in amplitude, but also will become modulated with the running speed when wear progresses (that is, 1X RPM sidebands will later appear above and below these natural frequencies). On the other hand, these sidebands can often be spaced at bearing defect frequencies themselves (BPFO, BPFI, BSF, etc.).

Note that the defects themselves in Stage 2 may not yet be readily visible to the naked eye. There should be only a slight increase in bearing noise and its temperature should still be roughly normal. Notice that bearing defect frequencies will not likely yet be visible in the velocity spectrum. However, acceleration spectra may now begin to pick up harmonics of defect frequencies, particularly if a log amplitude scale is employed. Still, at this stage, bearing defect frequency response will normally be erratic.

Figure 6.09U shows a classic machine which has advanced into Stage 2 of the failure scenario. In this case, Figure 6.09U pictured spectra taken from the inboard pump bearing of a 1790 RPM chilled water pump. On Figure 6.09U are shown trends of the overall spike energy, a cascade (or waterfall) diagram showing the historical spectra which have been taken on this machine, as well as the latest spectrum itself which vividly shows the problem. In the case of the spike energy trend, note that the overall levels have advanced from as low as .163 up to 5.14 gSE. In the case

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of the cascade diagram, note that no bearing component natural frequencies were evident on spectra taken up through March 29, 1993. However, they made their initial appearance during the May 20, 1993 survey and were surrounded by 1X RPM sidebands with the amplitude of the probable natural frequency at about .023 in/sec. Note that on July 21, the apparent natural frequency located at about 54,500 CPM grew from .023 up to .036 in/sec. Then, on the survey on October 7, the bearing advanced late into Stage 2 as evidenced by the growth of the probable 54,500 CPM natural frequency up to .071 in/sec; combined with the growth and the number of 1X RPM sideband families surrounding this frequency. Also, possible bearing frequencies were beginning to show as evidenced by noninteger running speed multiples at approximately 12.2X RPM (21,800 CPM) and 15.3X RPM (27,250 CPM) (although the bearing model number could not be confirmed at this time).

Stage 3 (Approximately 1% to 5% L_{10} Life Remaining):

Note that each of 3 progressive events are documented for Stage 3 in Figure 6.09P (shown as A, B and C). The letters shown below the horizontal axis of the Stage 3 spectrum in Figure 6.09P correspond to peaks which appear during these 3 progressive events. For the first time, bearing defect frequencies associated with faults in the inner race (BPFI), outer race (BPFO), rolling elements (BSF) and/or cage (FTF) appear in the velocity spectrum (Event A). Later on in Stage 3, harmonics of these bearing defect frequencies then will appear as slight wear progresses around the periphery of the raceways and/or faults appear on more than one rolling element (Event B). Normally, the defect frequencies themselves will not be modulated by running speed when they first appear (unless the bearing is fairly heavily loaded in which case BPFI will often be surrounded by 1X RPM sidebands when it first appears since the inner race defect will rotate in and out of the load zone). When deterioration progresses, 1X RPM sidebands will surround the bearing defect frequencies. Spike energy (or equivalent) will continue to grow, doubling or tripling in amplitude. At the end of Stage 3, not only will 1X RPM sidebands appear around bearing defect frequencies, but more sideband families will appear around the bearing component natural frequencies (Event C).

A word of caution should be taken at this point. When the bearings approach the conclusion of Stage 3, the rate of wear becomes highly unpredictable. In any case, it will be well into the exponential part of the failure curve (Figure 6.09N). How much longer the bearing lasts will largely depend on its lubrication, temperature, cleanliness and dynamic loads being imposed upon it by vibration forces from unbalance, misalignment and so forth. At this point, there will be a noticeable change in sound level and frequency and a slight increase in bearing housing temperature. Addition of lubricant at this point may temporarily lower spike energy and possibly have some effect in reducing vibration. However, since wear is the problem at this point, both vibration and spike energy will return, normally within 12 to 24 hours (the lubricant addition at this point could hasten failure if the bearings already were over-lubricated).

Figures 6.09V and 6.09W show a spectrum and spike energy trend plot, respectively, taken from a pump bearing with a classic Stage 3 bearing failure underway. Notice that the spike energy increased from .271 to .944 gSE between the May 24 and August 23 measurements exceeding the alarm of .750 gSE (Figure 6.09W). Then, looking at the velocity spectrum of Figure 6.09V, each of 4 harmonics of inner race frequency are present (BPFI = 7.44 X RPM = 13,200 CPM). Note the harmonics at 26400, 39600 and 52800 CPM. Also note the 1X RPM sidebands which have appeared around 2BPFI and 3BPFI. Looking at both the velocity spectral data and spike energy data, the plant would be well advised to replace this bearing right away even though amplitudes of all bearing frequencies are below .10 in/sec.

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FIGURE 6.09U
COMMENCEMENT AND DETERIORATION OF FAILURE STAGE 2 ON A PUMP BEARING

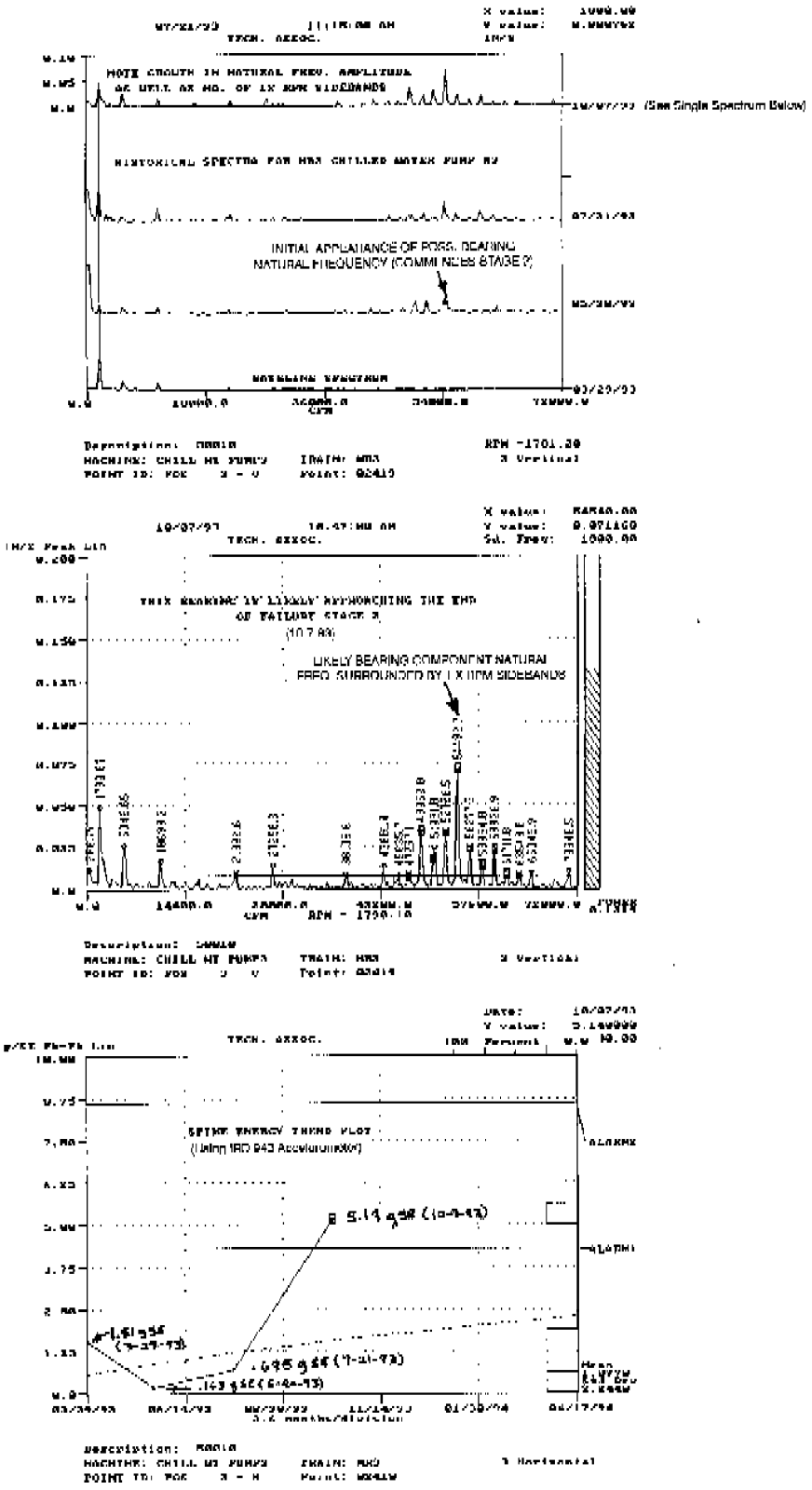
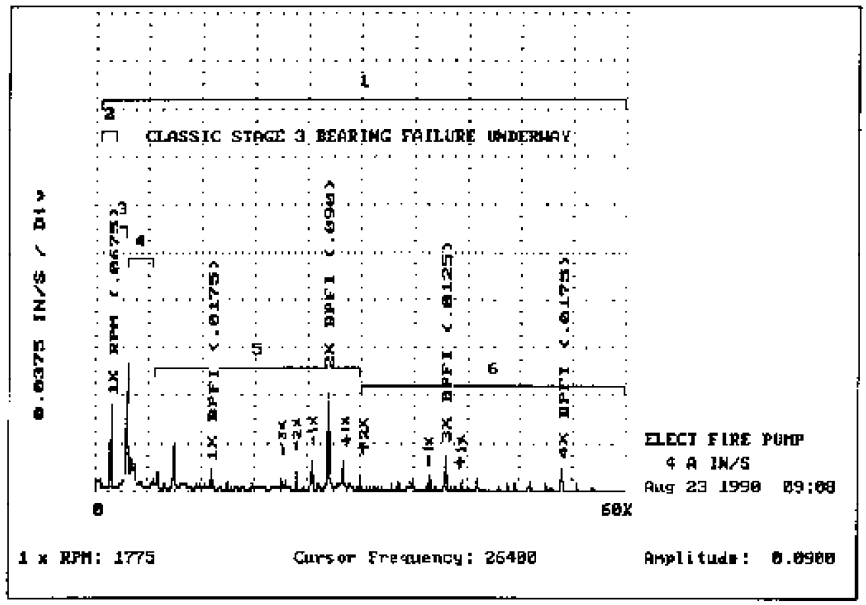

NOTE GROWTH IN NATURAL FREQ. AMPLITUDE AS UNIT GOES NO. OF 1X RPM SIDEBANDS
(See Single Spectrum Below)
HISTORICAL SPECTRA FOR HRS CHILLED WATER PUMP #2
INITIAL APPEARANCE OF POSS. BEARING NATURAL FREQUENCY (COMMENCES STAGE 2)
BASELINE SPECTRUM
TECH. ASSOC.
LIKELY BEARING COMPONENT NATURAL FREQ. SURROUNDED BY 1 X RPM SIDEBANDS
SPIKE ENERGY TREND PLOT
(Using IRD 843 Accelerometer)

FIGURE 6.09V



2% x 0.500 = 0.0100 1 x RPM: 1775.00

Machine	Pos	Dir	Units	Date
ELECT FIRE PUMP	4	A	IN/S	Aug 23 1990 09:08

Frequency	1 x RPM	Amplitude	
150	0.08	0.0100	
600	0.34	0.0100	
1800	1.01	0.0675	
3600	2.03	0.1000	
4050	2.28	0.0250	
4500	2.54	0.0200	
6600	3.72	0.0100	
7200	4.06	0.0150	
8850	4.99	0.0350	
13200	7.44	0.0175	← BPFI
21000	11.83	0.0100	- 3X RPM
22800	12.85	0.0150	2X RPM
24600	13.86	0.0225	- 1X RPM
26400	14.87	0.0900	← 2X BPFI
28200	15.89	0.0225	+ 1X RPM
30000	16.90	0.0125	+ 2X RPM
37800	21.30	0.0125	
39600	22.31	0.0275	← 3X BPFI
41400	23.32	0.0100	
43200	24.34	0.0100	
52800	29.75	0.0175	← 4X BPFI

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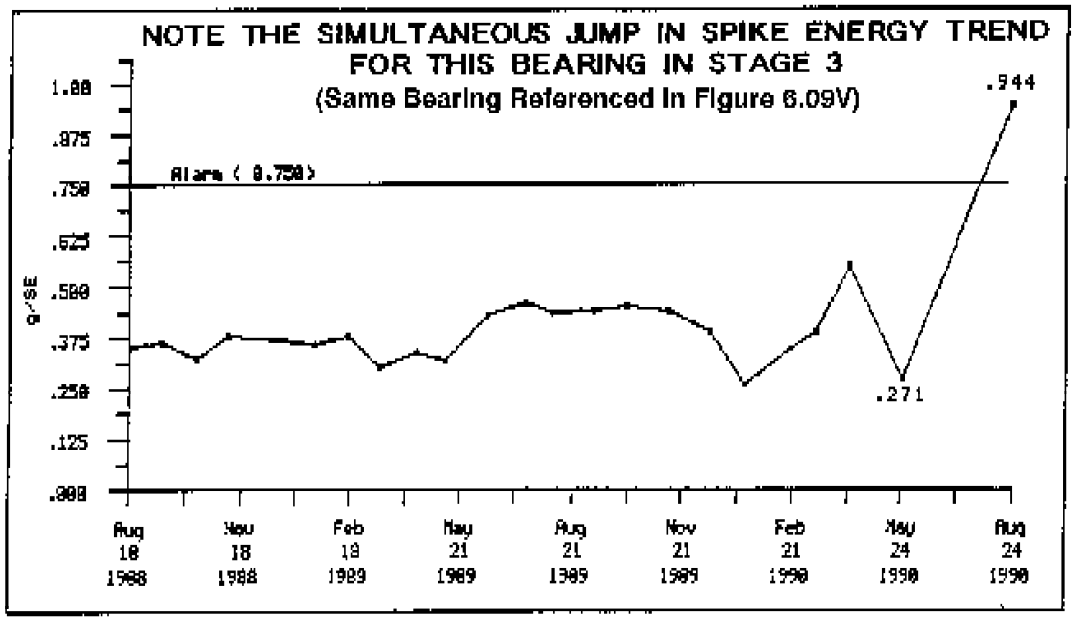
FIGURE 6.09W

[redacted] Trend Graph Nov-26-90



ELECT FIRE PUMP 4 H .750 g/SE

IRD Machanalysis Data Analysis System

Machine ELECT FIRE PUMP Pos 4 Dir H Units g/SE Alarm .750

Ampl	Insp Code	Date mm/dd/yy	Time	Ampl	Insp Code	Date mm/dd/yy	Time
.353	0	08-18-88	9:25	.442	0	09-07-89	8:51
.361	0	09-13-88	0:07	.449	0	10-05-89	8:50
.320	0	10-11-88	9:18	.442	0	11-09-89	10:00
.378	0	11-07-88	9:19	.390	0	12-14-89	15:38
.368	0	12-20-88	10:34	.261	0	01-11-90	8:52
.355	0	01-17-89	9:10	.347	0	02-19-90	8:40
.380	0	02-14-89	8:51	.388	0	03-12-90	8:58
.302	0	03-14-89	8:45	.549	0	04-09-90	7:54
.338	0	04-13-89	8:03	.271	0	05-24-90	7:51
.316	0	05-09-89	9:18	.944	0	08-23-90	9:08
.429	0	06-13-89	9:17				
.460	0	07-13-89	9:20				
.434	0	08-03-89	9:25				

Page 1

Machine Report (Latest Data) for: ELECT FIRE PUMP

Date	Time	Pos	Dir	Ampl	Units	Change	Alarm	Inspection
Aug-23-90	09:06	1	A	.150	IN/S	-6%	.300	
Aug-23-90	09:06	1	H	.129	IN/S	48%	.300	
Aug-23-90	09:06	1	H	.329	g/SE	186%	.600	
Aug-23-90	09:06	1	O	.117	IN/S	------%	.300	
Aug-23-90	09:06	1	V	.077	IN/S	8%	.300	
Aug-23-90	09:07	2	A	.157	IN/S	-15%	.300	
Aug-23-90	09:07	2	H	.127	IN/S	14%	.300	
Aug-23-90	09:07	2	H	.333	g/SE	-1%	.600	
Aug-23-90	09:07	2	O	.129	IN/S	------%	.300	
Aug-23-90	09:07	2	V	.061	IN/S	-65%	.300	
Aug-23-90	09:07	3	A	.211	IN/S	41%	.300	
Aug-23-90	09:07	3	H	.180	IN/S	10%	.300	
Aug-23-90	09:07	3	H	.337	g/SE	42%	.600	
Aug-23-90	09:07	3	V	.189	IN/S	26%	.300	
Aug-23-90	09:08	4	A	.204	IN/S	-19%	.300	
Aug-23-90	09:08	4	H	.204	IN/S	-2%	.300	
Aug-23-90	09:08	4	H	*.944	g/SE	248%	.750	
Aug-23-90	09:09	4	V	.121	IN/S	-15%	.300	

18 lines this report

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Figure 6.09X shows evidence of another machine which has advanced well into Stage 3 as of its July 21, 1993 spectrum. An important point which should be made here is that this bearing passed completely through Stage 3 during the time interval between surveys of only 8 weeks (another reason why surveys should not be scheduled this far apart - the client elected to do this to "save money"). This data was taken on the inboard bearing of the motor which is a 6-pole induction motor operating at 1192 RPM and outfitted with NTN 2313 bearings. In this case, the outer race frequency was at about 5.19X RPM while the cage frequency (FTF) was at about .41X RPM. First, notice the great number of outer race frequency harmonics showing at the 2nd, 3rd, 5th, 7th, 8th and 9th harmonic outer race frequencies. In addition, what is probably of much greater concern are the great number of multiples of cage frequency sidebands surrounding these outer race frequencies, in addition to the appearance of the cage frequency itself (FTF). This bearing is in great jeopardy and should not be guaranteed to operate any length of time. Once the cage fails, catastrophic failure can occur quickly.

Figure 6.09Y is an example of a bearing where 4X BPFI initiated Stage 3 (rather than 1X BPFI). In fact, the fundamental inner race frequency (1X BPFI = 7.14X RPM) never did grow to significant amplitude. However, this figure showed the development of very serious Stage 3 bearing problems in this vertical pump operating at about 1787 RPM. Note especially the cascade plot which showed no real problems with the machine as of May 11, 1992. Then, the predominant bearing frequency which initially appeared was at 4X inner race frequency (4X BPFI) on June 9, 1992. Later, even more harmonics of inner race frequency appeared and grew to excessively high amplitudes (for bearing defect frequencies). In fact, on December 8, the amplitude at 4X BPFI was .267 in/sec and had a 1X RPM sideband up to .342 in/sec. Interestingly, note that the middle plot on Figure 6.09Y showed that during the next survey on January 13, 1993, the amplitudes at bearing frequencies actually dropped to .100 in/sec at 4X BPFI, and down to .081 in/sec at the 1X RPM sideband to the left of 4X BPFI which was excessively high on December 8. Note that it is not uncommon for amplitudes at bearing frequencies to actually decrease as the condition is truly getting worse. Note also the spike energy trend in Figure 6.09Y which finally showed about 10.5 gSE (excessive for measurements with an IRD 970 accelerometer). Although the plant had been strongly recommended to replace the bearing for several months, they were now told if they did not replace the bearing soon, it would soon take care of itself (catastrophically). The cascade spectrum in Figure 6.09Y shows the plant did finally replace the bearing and all vibration at bearing defect frequencies disappeared; and overall spike energy levels significantly dropped and have remained low since that time.

Stage 4 - Approaching Catastrophic Failure At Any Time (Approx. 1 Hour to 1% L_{10} Life Remaining):

The level of 1X RPM normally begins to grow for the first time throughout the bearing failure process in Stage 4, along with harmonics at 2X and 3X RPM. Spike energy levels typically actually drop and amplitudes of the higher bearing frequency harmonics and natural frequencies also normally drop. Many 1X RPM sidebands appear around bearing defect frequencies (indicating pronounced wear throughout the periphery of the bearing). There will now be a noticeable change in pitch of the bearing noise, and likely a significant increase in bearing housing temperature.

Later on in Stage 4, discernible bearing defect and component natural frequencies actually begin to disappear and are replaced by a random broadband high frequency "noise floor" which can extend far down into the spectrum, obliterating discrete frequency peaks. Finally, at the end of Stage 4, spike energy normally will decline again (as the bearing itself "disappears"); but, just prior to failure, the spike energy levels can grow dramatically up to 50 to 100 gSE just before final seizure. Thus, a bearing should never be allowed to operate in Stage 4 for no one knows when it will catastrophically fail.

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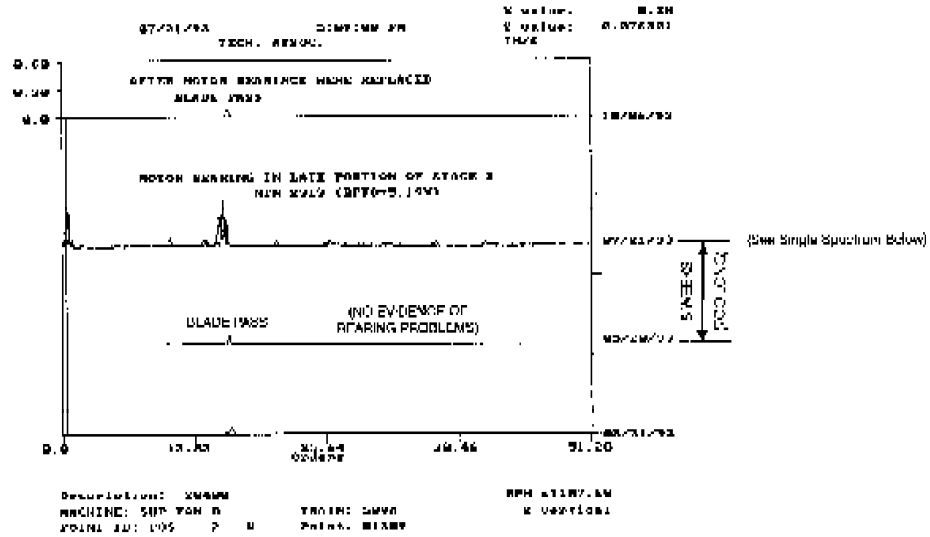
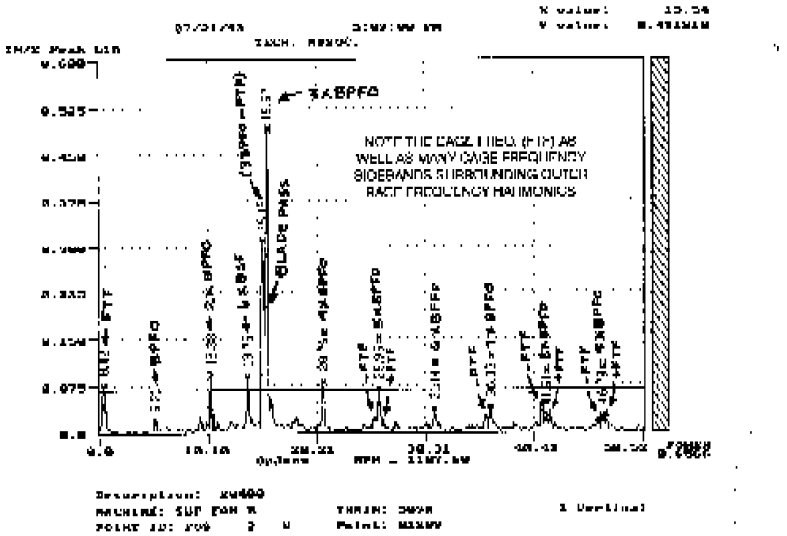
FIGURE 6.09X
EXAMPLE OF A BLOWER BEARING WHICH APPARENTLY SEEMED TO ADVANCE NEAR THE END OF FAILURE STAGE 3
(Due to too long a time interval between surveys to "Save Money")


AFTER MOTOR BEARINGS WERE REPLACED
MOTOR BEARING IN LATE PORTION OF STAGE 3
BLADE PASS
(NO EVIDENCE OF BEARING PROBLEMS)
STAGE 3
(See Single Spectrum Below)


1X BPFO
NOTE THE SIDEBANDS (FTF) AS WELL AS MANY CAGE FREQUENCY SIDEBANDS SURROUNDING OUTER RACE FREQUENCY HARMONICS
BLADE PASS

07/21/93
5:09:00 PM

Threshold Report

PLANT: PPG
Description: 20400
MACHINE: SUP FAN B
POINT ID: POS 2 - V
Units: IN/S Peak Lin
Threshold: 0.067000

TRAIN: 509A
Point: 01209
RPM: 1187.60

Frequency Hz	CPM	Order	Spect Amp	% Exceeded
8.34	500.6	0.42	0.076301	13.88
205.45	12327.2	10.38	0.092370	37.87
272.10	16326.3	13.75	0.090190	34.61
299.94	17996.3	15.15	0.312073	366.23
308.26	18495.3	15.57	0.491818	634.06
410.91	24654.4	20.76	0.081970	22.34
513.60	3.08E+4	25.95	0.067710	1.06

Power	Sync	Subsync	Non-Sync
0.056627	0.327053	0.081327	0.563543

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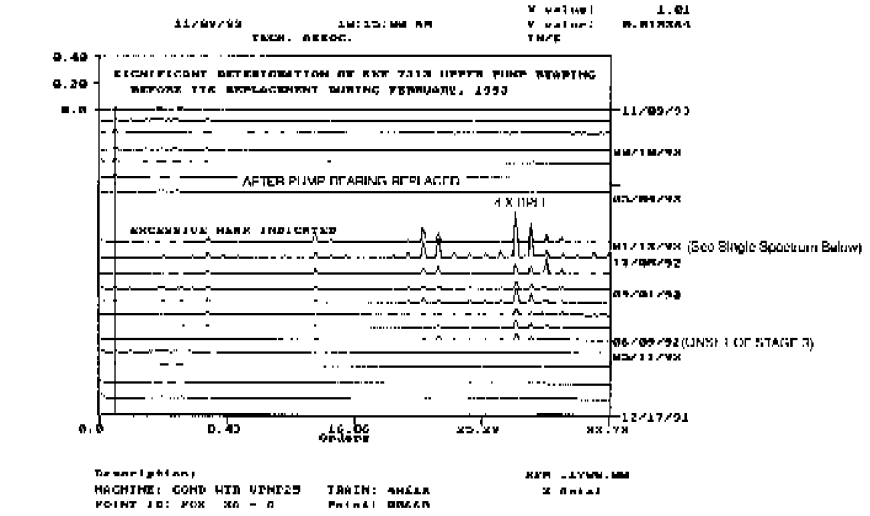
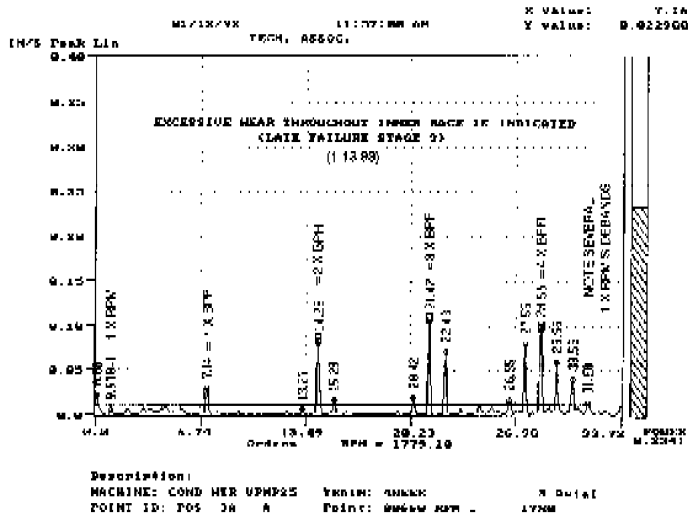
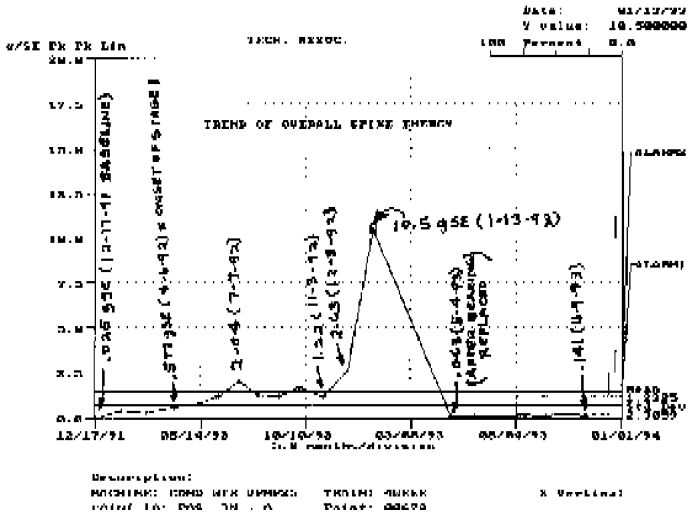
FIGURE 6.09Y
EXAMPLE OF A BEARING WHERE 4X BPFI INITIATED STAGE 3 (RATHER THAN 1X BPFI) AND LATER GENERATED MORE BPFI HARMONICS AS WELL AS 1X RPM SIDEBANDS WHEN ITS CONDITION DETERIORATED


SIGNIFICANT DETERIORATION OF NDE 7313 UPPER PUMP BEARING
BEFORE ITS REPLACEMENT DURING FEBRUARY, 1993
AFTER PUMP BEARING REPLACED
4 X (BPFI)
EXCESSIVE WEAR INDICATED
(See Single Spectrum Below)
12/17/91





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While most bearings track through the failure stage presented in Scenario A, some will take other paths. These will be outlined in Scenarios B through F which will follow.

SCENARIO B. CONTINUED DETERIORATION OF ONE PRONOUNCED FAULT ON A RACEWAY (Figure 6.09Z):

Normally, faults appear on one bearing component and then spread to the other components through the rolling elements due to the impact vibration and dynamic forces they generate. However, occasionally a single fault will appear on one or the other raceways and the bearing will continue to failure with very little effect on the other components. This may be due to possible metallurgical problems with the material of a raceway or other related problems. In these cases, the bearing will normally pass through the first 2 failure stages outlined in Scenario A. However, when in Stage 3, the single fault will predominantly excite only one race defect frequency which itself will generate little or no harmonics. Sidebands at 1X RPM will then appear around this singular defect frequency as shown in Figure 6.09Z (note sidebands about BPFI).

Later on, in Stage 4, the single defect frequency itself will typically increase in amplitude dramatically. In the author's experience, these frequencies have been seen to grow to 1.0 in/sec or more if the bearing has a localized fault as is the case of Scenario B (this rarely happens to the amplitude of any bearing frequency in Scenario A). Then, 1X RPM sidebands will grow to amplitudes much higher than normal. Sometimes, more than one family of sidebands will appear around this frequency (as shown in Stage 4 of Figure 6.09Z). Late in Stage 4, harmonics of this defect frequency will appear and finally, defect frequencies for the remaining bearing components do also as seen in the figure. At this point, the amplitude at 1X RPM and lower harmonics is affected. If the bearing is not replaced, it often will almost literally fall apart when removed, and there will be one pronounced defect of considerable depth on at least one raceway, surrounded by general wear throughout the remainder of the bearing.

SCENARIO C. CONTINUAL WEAR THROUGHOUT THE PERIPHERY OR LOAD ZONE OF ONE RACEWAY (Figure 6.09AA):

Figure 6.09AA illustrates this failure mode. Note that it passes through Stage 1 just as in Scenario A with the appearance only of increased amplitudes of overall ultrasonic parameters (Spike Energy, HFD, etc.). However, then one of the first frequencies to appear will be some harmonic of a race frequency (for example, the 4th or 5th harmonic outer or inner race frequency similar to that shown in Figure 6.09AA). In these cases, it is not yet clear why a harmonic defect frequency first responds before the bearing component natural frequencies. One possibility is that a system natural frequency (rotor, frame, etc.) may be located nearby a particular defect frequency harmonic (such as 4BPFO as shown in Figure 6.09AA). In this case, any excitation of this resonant frequency would generate much more response from this peak located close to the natural frequency than that from the other harmonics of the defect frequencies.

Later on in this failure mode, Stage 3 shows the appearance of more race frequency harmonics (in this case, 3BPFO and 7BPFO, in addition to 4BPFO). Another interesting thing happens in this particular failure mode. 1X RPM sidebands often do not appear. Still, the wear is becoming serious and is progressing around the periphery of the raceway. Later, in the final stages (shown in Stage 4), even more race frequency harmonics appear. They grow in amplitude, but still often are not surrounded by 1X RPM sidebands. Even though the height of any singular race defect frequency harmonic may only be .03 to .05 in/sec, a bearing generating this many harmonics of defect frequencies will normally show excessive wear which borders on bearing seizure.

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FIGURE 6.09Z
CONTINUED DETERIORATION OF ONE PRONOUNCED FAULT ON A RACEWAY (Scenario B)

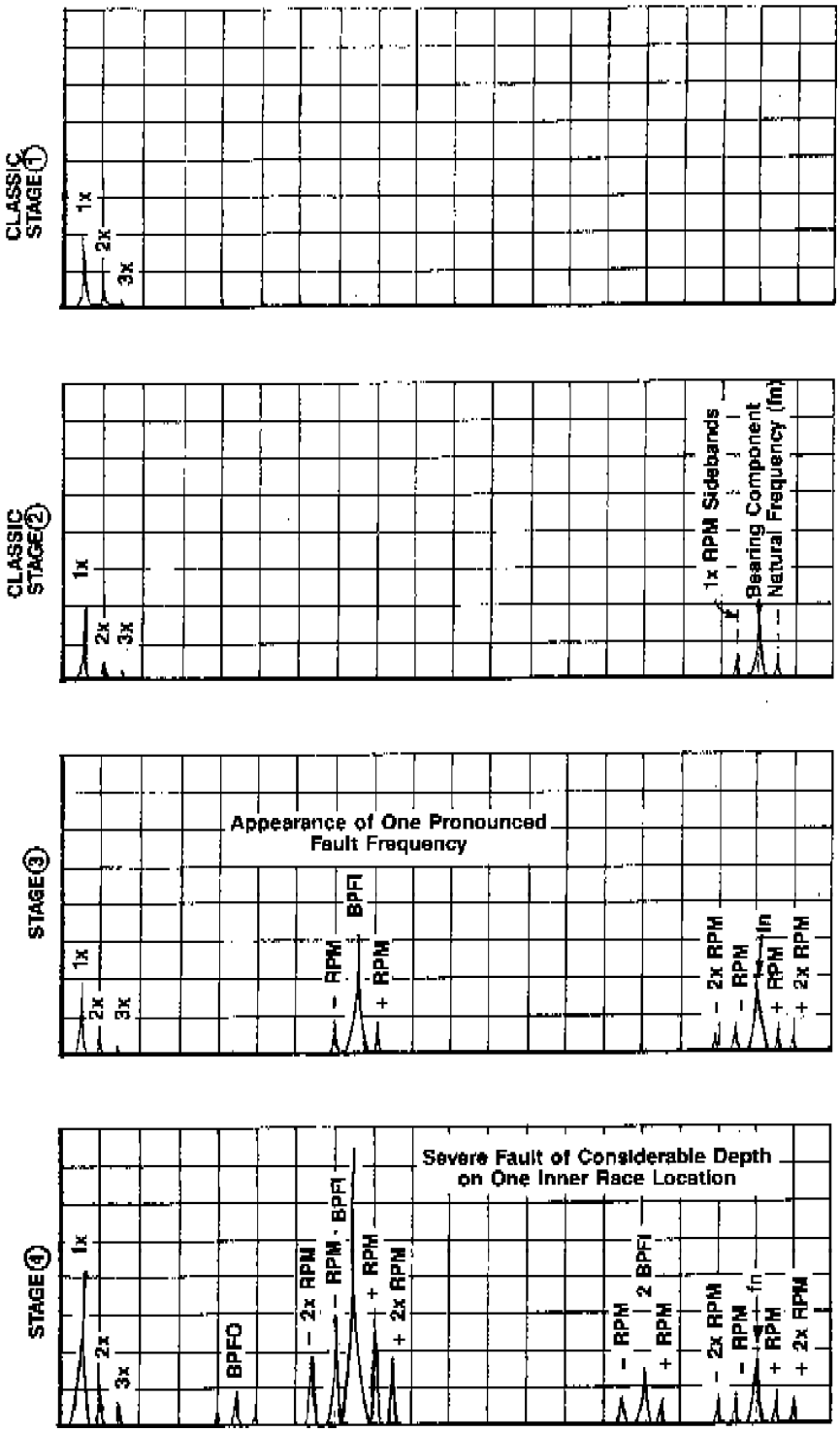

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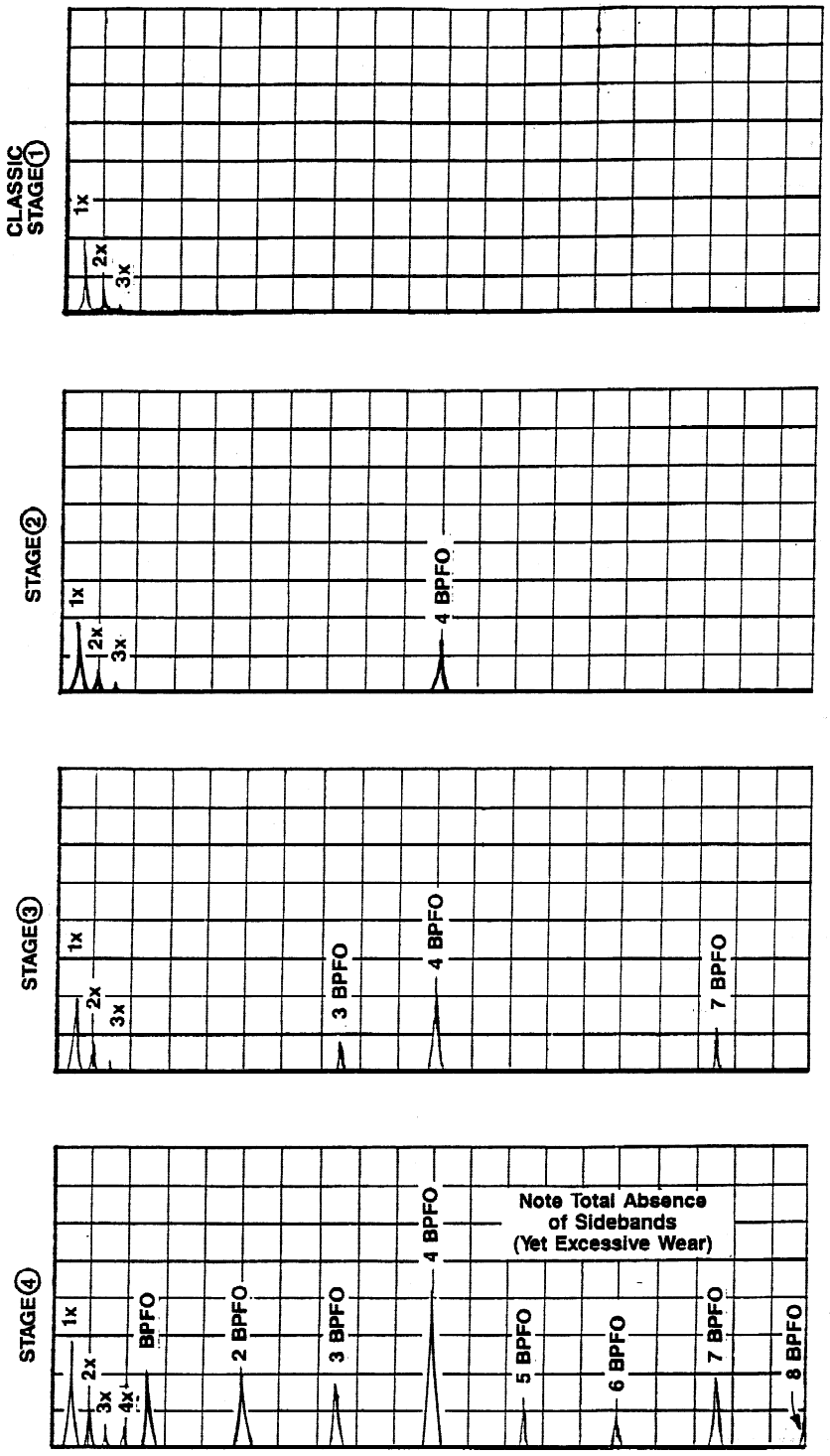
FIGURE 6.09AA
CONTINUAL WEAR THROUGHOUT THE PERIPHERY OR LOAD ZONE OF ONE RACEWAY (Scenario C)



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SCENARIO D. DEVELOPMENT OF A SERIOUS FAULT FREQUENCY ACTING AS A SIDEBAND RATHER THAN AS A FUNDAMENTAL (Figure 6.09BB):

Scenario D illustrates another way impending failures can be detected by signature analysis. Not only can the rolling element bearing defect frequencies appear as fundamentals or harmonics, they also can act as sidebands around other forcing frequencies. For example, when faults appear on balls or rollers, they often will generate a cage frequency (FTF) since the rolling elements impact the cage regardless of the rotation. However, the cage frequency itself seldom appears at its fundamental of about .35 to .45X RPM. Instead, it will more often modulate other frequencies causing sidebands spaced at cage frequency (FTF). When several defects are present, some will be identified from their fundamental frequency, but others will act at these Sum and Difference Frequencies. This is illustrated in Figure 6.09BB. Note the appearance of the cage frequency (FTF) acting as a sideband about a race frequency (BPFI in this case) in Stage 3. Note that when deterioration continues as shown in Stage 4, advanced faults will likely be present throughout the bearing. Note the multiple families of cage frequency sidebands around the 1st and 2nd harmonic inner race frequency and the 1X RPM sidebands around the ball spin frequency (BSF) and the bearing component natural frequency (f_n) in this example. When this occurs, the life of the bearing is in great jeopardy.

SCENARIO E. CONDITION DETERIORATION ENDING EITHER WITH SEVERE MECHANICAL LOOSENESS OR THE BEARING TURNING ON THE SHAFT (Figure 6.09CC):

Figure 6.09CC tracks the onset of a much more subtle bearing/shaft failure evidenced by the appearance of more and more running speed harmonics. This can eventually end up with either severe mechanical looseness or the bearing turning on the shaft (it is not possible at this point to differentiate between the two as they end up with almost identical spectra). Note the additional running speed harmonics appearing in Stages 2 and 3, with even more generated in Stage 4. However, note the absence of any bearing defect frequencies. In this case, the problem can be mechanical looseness, or it can be a problem of much greater severity - that is, a bearing turning on the shaft. In neither case would significant wear be expected in the bearing itself. However, the danger is the fact that significant damage can be taking place on the shaft due to a bearing turning on it and yet, very little vibration amplitudes may be occurring (commonly less than .05 in/sec). Therefore, if a signature approaching spectra shown in Stages 3 or 4 is captured as a baseline, it could be a moderate problem (mechanical looseness), or it could be a very serious problem (bearing turning on shaft). In these cases, it would be a good idea to lift up on the shaft (with the machine stopped) to determine the amount of play to help guide which problem is dominant and what steps should be taken.

SCENARIO F. DEVELOPMENT OF EXCESSIVE 1X RPM MODULATION ABOUT RACE FREQUENCIES ENDING UP WITH A WHOLE SERIES OF RUNNING SPEED HARMONICS CAUSED BY EXCESSIVE FATIGUE SPALLING ON RACEWAYS (Figure 6.09DD):

Figure 6.09DD illustrates this problem involving excessive fatigue spalling on raceways. When this occurs, the bearing will normally pass through the first 2 stages normally, but will generate considerably more running speed sidebands about race defect frequencies in Stage 3. When this occurs, it can eventually cause a spectrum like that in Stage 4 - that is, a fairly high once per revolution vibration, followed by a whole series of running speed harmonics as the 1X RPM sidebands pretty well take over this spectrum. In this case, the noise floor will also normally lift up during the final stages before failure as shown in Stage 4 of Figure 6.09DD. At this point, even though 1X RPM might be of considerable magnitude, balancing will not really help the problem (and may not be even possible at this time).

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FIGURE 6.09BB
DEVELOPMENT OF A SERIOUS FAULT FREQUENCY ACTING AS A SIDEBAND RATHER THAN A FUNDAMENTAL (Scenario D)

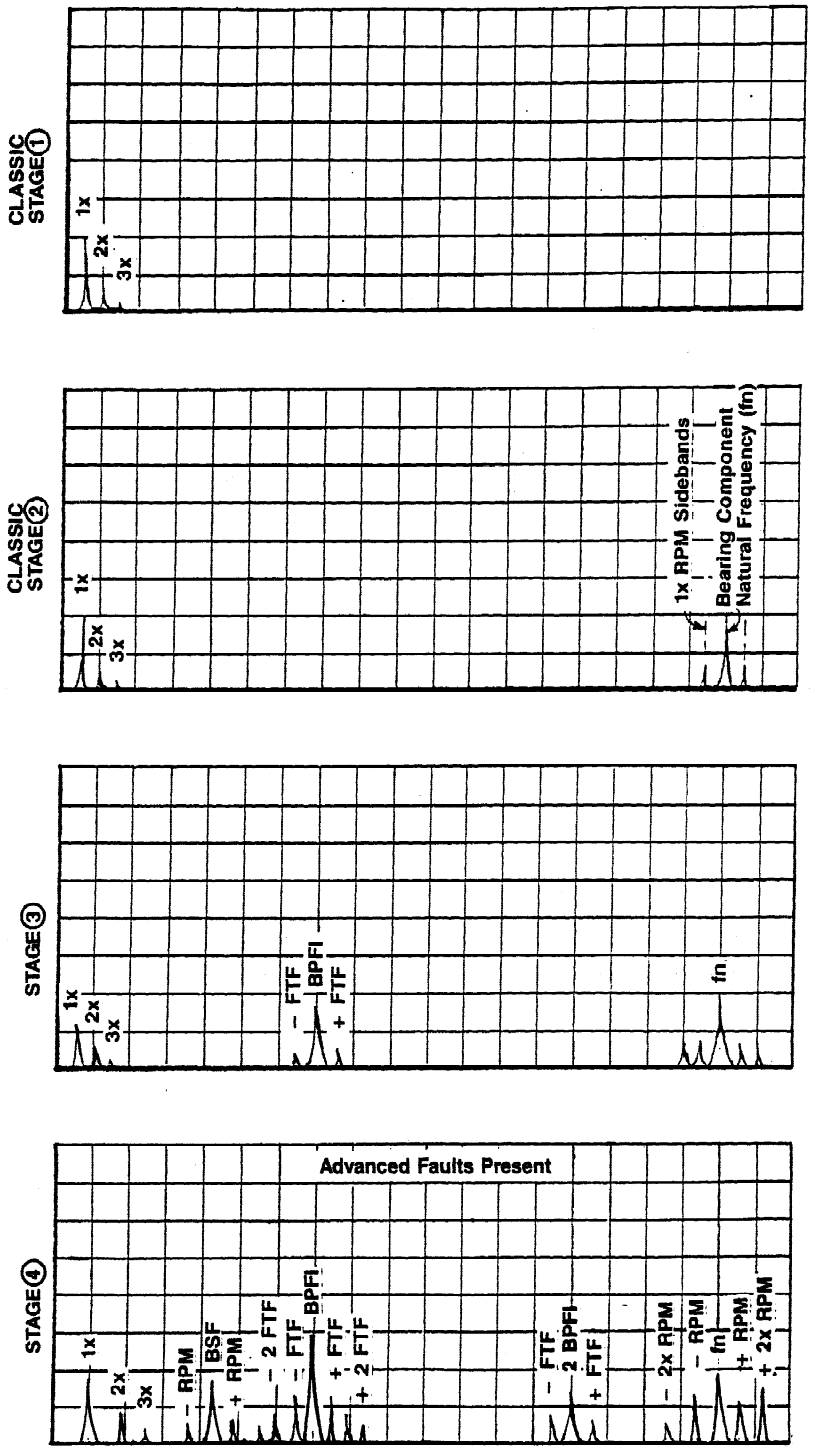

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FIGURE 6.09CC
CONDITION DETERIORATION ENDING EITHER WITH SEVERE MECHANICAL LOOSENESS OR THE BEARING TURNING ON THE SHAFT
(Scenario E)

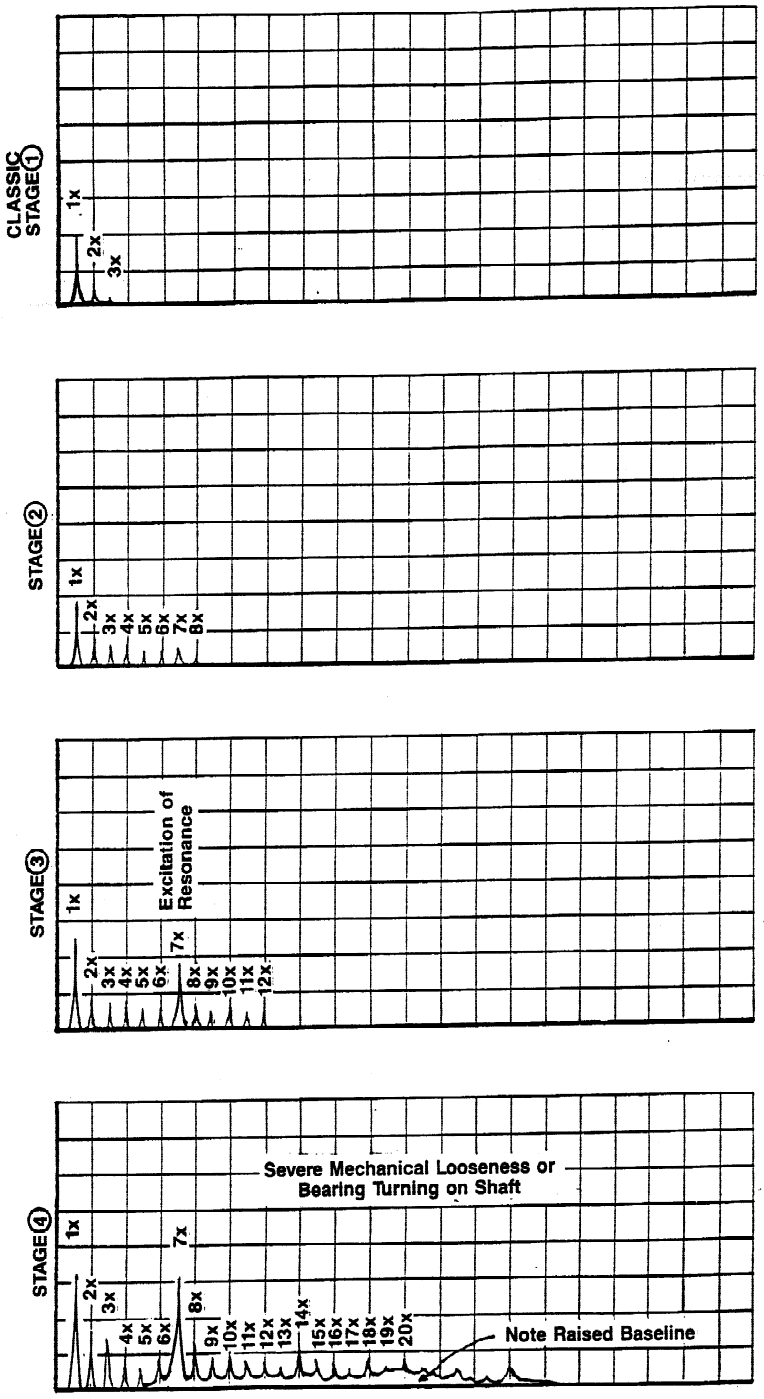

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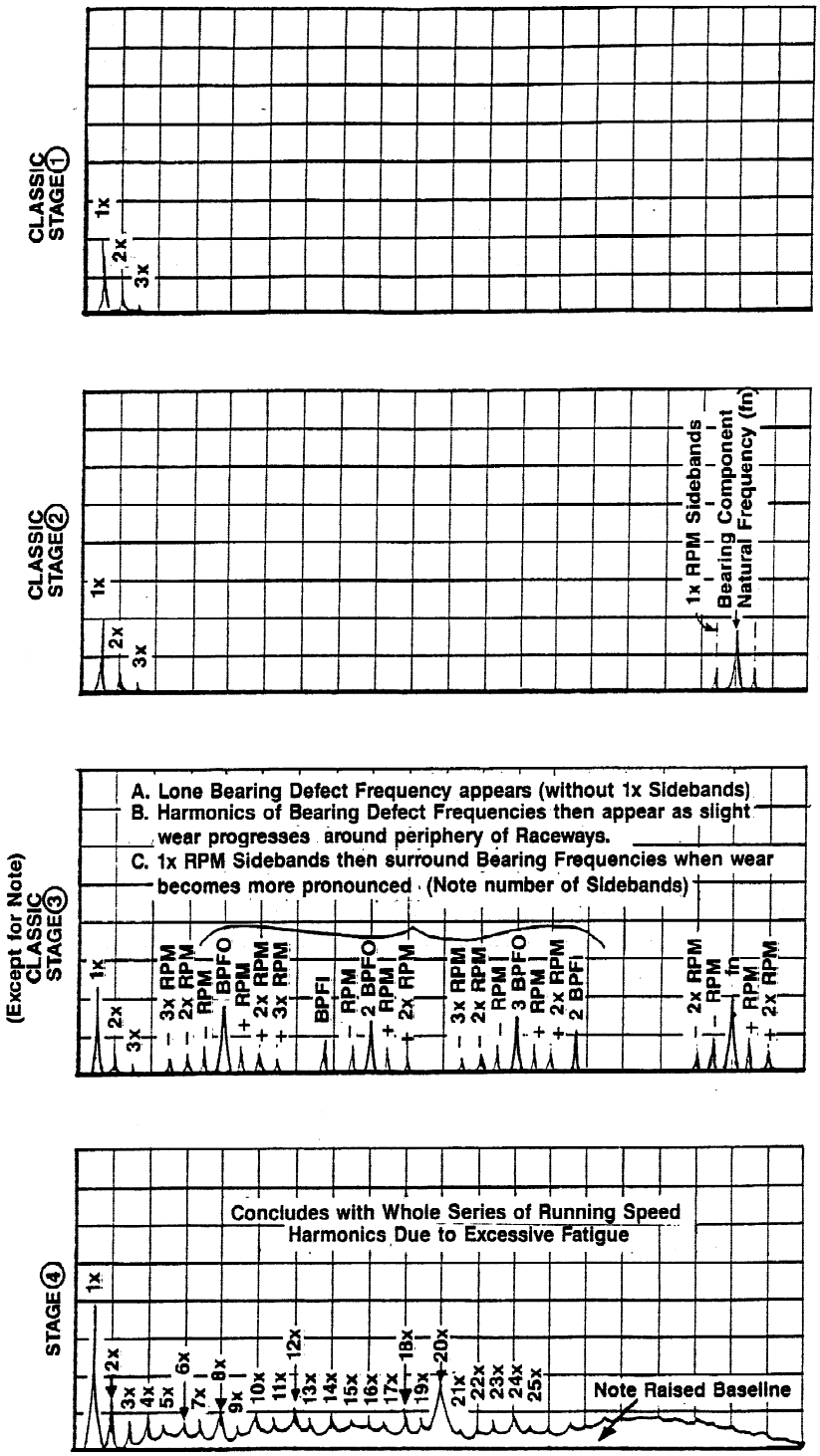
FIGURE 6.09DD
DEVELOPMENT OF EXCESSIVE 1X RPM MODULATION ABOUT RACE FREQUENCIES ENDING UP WITH RUNNING SPEED HARMONICS CAUSED BY EXCESSIVE FATIGUE SPALLING ON RACEWAYS (Scenario F)



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6.094 WORD OF WARNING CONCERNING INSTRUMENTS AND TRANSDUCER MOUNTINGS

A. HOW 8 BIT DATA COLLECTORS CAN MISS POTENTIALLY SERIOUS BEARING PROBLEMS:

The chart on Figure 6.09EE helps demonstrate the difference between an 8 bit and a 12 bit data collector. Importantly, the dynamic range of an 8 bit instrument is about 48 dB versus about 72 dB for a 12 bit instrument as shown in Figure 6.09EE. This 24 dB difference translates into a sensitivity difference of almost 16 times - that is, 12 bit data collectors are 16 times as amplitude sensitive as are their 8 bit counterparts. Experience has shown that 8 bit data collectors are not sufficiently sensitive to detect many rolling element bearing problems until the very final stages. Figure 6.09FF and the following anecdote help illustrate why this is so significant:

Charlie and Joe were two mechanics who decided to go into the vibration analysis business together. One day, opportunity struck when they travelled down to the local K-Mart there in Hertzville where they laid their eyes on some fancy, new "vibration rulers". They noticed two types were available. Each ruler ranged in length from 0 to 6 inches, but the "12 bit" ruler was delineated in 1/16 inch increments while the "8 bit" ruler was divided only into full 1 inch increments (see Figure 6.09FF). Now, Charlie thought he knew a good bargain when he saw one and he could see no reason why he should pay $99 for that fancy "12 bit" ruler when he reasoned he could get by just fine with the "8 bit" model at only $10. In fact, Charlie thought "Why, with all those little marks, they would probably just confuse me anyhow".

On the other hand, Joe went ahead and invested the extra $89 since he thought he just might need that "little extra accuracy" sometime, and also reasoned that this "12 bit" ruler was 16 times as sensitive as was the "8 bit" model so it "might" be worth it sometime.

The next day, Charlie and Joe went on their first job with their shiny new "vibration rulers". On the first machine, they noticed it was "shaking and baking" quite a bit, so they paid close attention. Figure 6.09FF shows the vibration peaks that were present, each of which was measured by Charlie's "8 bit" and Joe's "12 bit" vibration rulers. Figure 6.09FF shows that Charlie measured a big spike at 1X RPM and claimed it was 6 inches high. Joe said "No, not quite, it's only 5 3/4 inches high". Charlie retorted, "Big deal! What's the difference anyhow?" Joe had to reply "Well, very little I guess", and Joe began to worry whether or not the extra "5 Bucks" was really worth it.

However, Charlie then said "Well, that's it. I guess we're ready to go to the next machine". But Joe replied, "Wait a minute, Charlie. We need to write down the measurements on those other 3 peaks, too". Charlie answered "What peaks?" Joe replied, "You know, those other 3 peaks out there with the 'whatcha-ma-call-its' surrounding that bigger peak on each side". Charlie looked at his ruler again and said, "Joe, I'm telling ya - there just ain't nothing out there!" Joe abruptly answered "Well Charlie, I'm telling you - there's a bearing frequency out there 5/16 inch high and there's 2 sidebands on each side of it that are 1/8 inch high each - this machine has a bearing problem!"

About this time, Charlie's heart began to sink as he realized he just blew "10 Bucks" and must now spend $89 more when he recalled the little sign accompanying the vibration ruler display saying "Blue Light Specials On Sale Today - No Returns". On the other hand, Joe felt pretty good when he left for home that day since his first customer gave him an extra $250 bonus for uncovering what could have been a serious problem had they not caught it in time on this "No.1 Widget Fabricator" which was their most critical machine in the plant!

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Now that Joe and Charlie have helped illustrate the difference between an 8 bit and 12 bit instrument, we can examine what happened on a forming fan at a major industrial client when measurements were simultaneously taken with both an 8 bit instrument (IRD 818) and a 12 bit instrument (IRD 890) on the same bearing housing, using the same IRD 970 accelerometer connected to each instrument (initially connected it to the 890; then the 970 was left in place, but the connector at the data collector was removed from the 890 and attached to the 818). Note that the cascade spectrum shown in Figure 6.09GG shows that measurements had been taken for several surveys with the IRD 818. Then, when Technical Associates visited the plant with an IRD 890, an abruptly different signature was noticed by our analyst as per the 12/3/90 spectrum in Figure 6.09GG. This was reported to the plant and a decision was made to conduct the simultaneous measurement test at first opportunity.

On 12/5/90, this simultaneous measurement test was performed. First the IRD 890 captured the data at 13:51; then the transducer cable connector was removed from the 890 and connected to the 818 (leaving the accelerometer in place). The 818 captured the spectrum at 13:52 which is also shown in Figure 6.09GG. The important difference between the spectra captured by the 8 bit and 12 bit instruments is seen by referring to Figure 6.09GG and the discussion on "Failure Stage 3". Note that when a pair of sideband frequencies appear around a bearing defect frequency, they indicate further bearing deterioration. In the case of the 12 bit spectrum on Figure 6.09GG, note the presence not only of a single pair of sidebands, but 2 fully developed sideband families and a third on the way. Although this 12 bit spectrum does not indicate imminent failure, particularly due to the absence of any other bearing frequencies or bearing frequency harmonics, the multiple families of sidebands that were totally missed by the 8 bit instrument do suggest that this machine deserves close attention.

This is just one example of important information that has been detected by 12 bit instruments that has been missed by 8 bit models. This strongly points out the need to move up from 8 bit data collector models as soon as possible.

DECIBEL CONVERSION TABLE

$dB = 20 \text{ LOG } (V_i/V_O)$

$V_i = V_O \text{ LOG}^{-1} \left(\frac{dB}{20}\right)$

dB	Ratio	Dynamic Range			
		A/D # Bits Used	# Of Bits Value	Largest Number	dB Range
100	100,000:1	1	1	2	6
90	31,623:1	2	2	4	12
80	10,000:1	3	4	8	18
70	3,162:1	4	8	16	24
60	1,000:1	5	16	32	30
50	316:1	6	32	64	36
48	251:1	7	64	128	42
46	200:1	8	128	256	48
40	100:1	9	256	512	54
34	50:1	10	512	1024	60
26	20:1	11	1024	2048	66
20	10:1	12	2048	4096	72
14	5:1	13	4096	8192	78
12	4:1	14	8192	16,384	84
10	3.16:1	15	16,384	32,768	90
8	2.51:1	16	32,768	65,536	96
6	2:1				
5	1.78:1				
4	1.58:1				
3	1.41:1				
2	1.26:1				
1	1.12:1				
0	1:1				

$\frac{4096}{256} = \frac{16}{}$ (24dB)

1 Part in 100 = 40dB
1 Part in 200 = 46dB (8 BIT)
1 Part in 1000 = 60dB
1 Part in 2000 = 66dB
1 Part in 4000 = 72dB (12 BIT)

12 BIT A/D INSTRUMENTS ARE 16 TIMES AS SENSITIVE AS ARE 8 BIT INSTRUMENTS!

FIGURE 6.09EE
8 BIT VERSUS 12 BIT INSTRUMENTS

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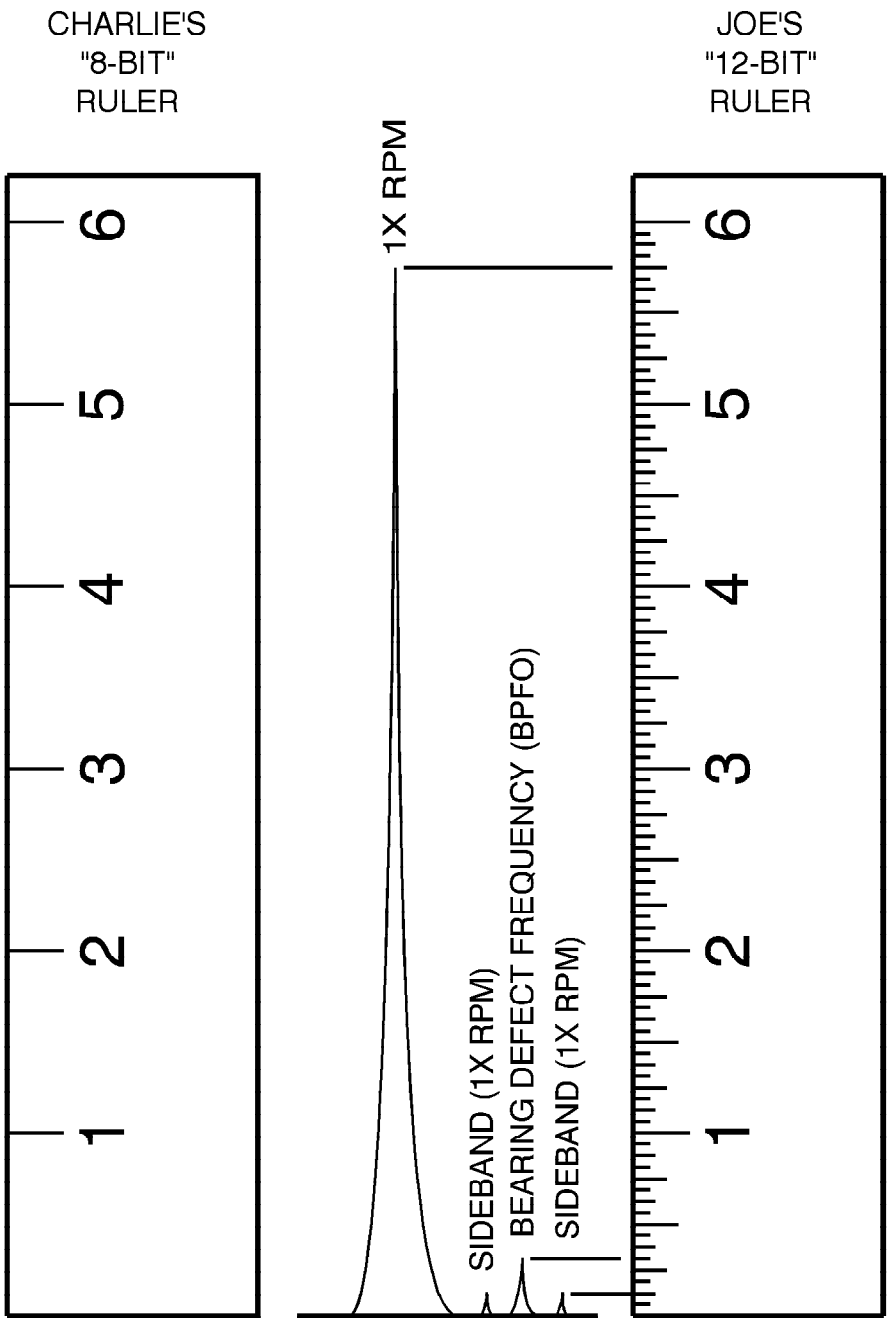
FIGURE 6.09FF
COMPARISON OF AMPLITUDE MEASUREMENTS DETECTED BY CHARLIE'S "8-BIT" AND JOE'S "12-BIT" VIBRATION RULER



CHARLIE SAYS:
1X RPM = 6" HIGH
BEARING FREQ. IS NOT THERE!
SIDEBANDS NOT THERE EITHER!

JOE SAYS:
1X RPM = 5 3/4" HIGH
BEARING FREQ. = 5/16" HIGH
SIDEBANDS = 1/8" HIGH EACH

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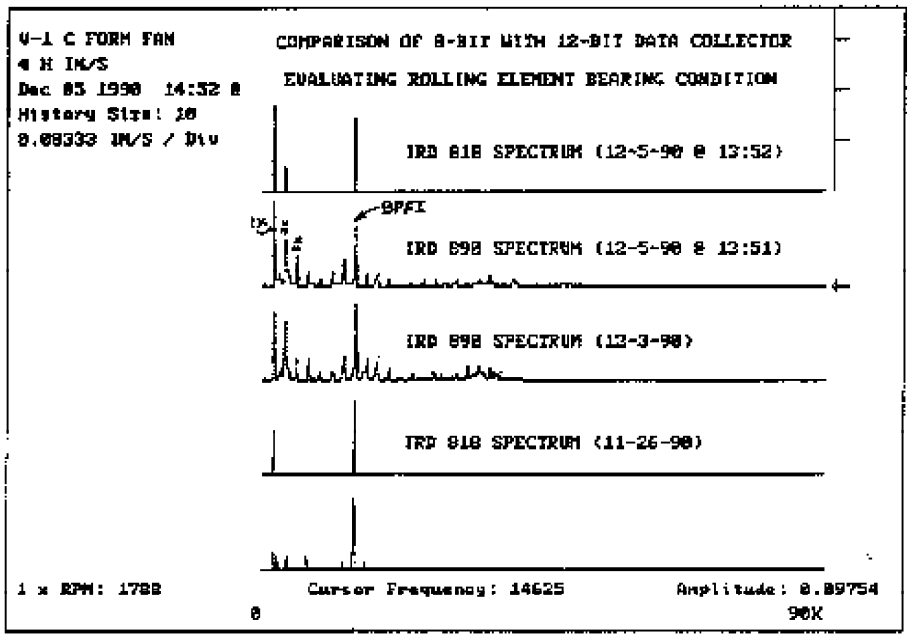
FIGURE 6.09GG



002% x 0.5000 = 0.01000 1 x RPM: 1788.00

Machine	Pos	Dir	Units	Date
V-1 C FORM FAN	4	H	IN/S	Dec 05 1990

Frequency	1 x RPM	Amplitude		Frequency	1 x RPM	Amplitude
1800	1.01	0.14249		40050	22.40	0.01001
2700	1.51	0.02002				
3600	2.01	0.07752				
5400	3.02	0.04996				
7200	4.03	0.02248				
9000	5.03	0.01001				
11025	6.17	0.02002	← BPFI - 2X			
12825	7.17	0.04750	← BPFI - 1X			
14625	8.18	0.09754	← BPFI			
16425	9.19	0.02248	← BPFI + 1X			
18225	10.19	0.02248	← BPFI + 2X			
20025	11.20	0.01247	← BPFI + 3X			
25650	14.35	0.01001				
27450	15.35	0.01001				
30825	17.24	0.01247				
34425	19.25	0.01247				
36225	20.26	0.01501				

DETECTED BY 12-BIT IRD 890

T - Threshold adjust Q - Quit

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B. IMPACT OF TRANSDUCER MOUNTING ON DETECTING ROLLING ELEMENT BEARING PROBLEMS:

Possibly, even more important and prevalent a problem than the "8 bit/12 bit" instrument performance question is the mounting of transducers when periodic vibration surveys are taken. Figure 6.09HH (provided by Bruel & Kjaer) illustrates this key point by comparing the frequency response for seven different mountings of a B&K 4367 accelerometer (whose characteristics closely resemble the standard accelerometers in use today by most data collector vendors). Notice how much the "maximum useful frequency" drops with different mountings as one moves from Type 1 through Type 7. For example, when the accelerometer is stud mounted on a thin film of silicon grease (Type 1), the mounting natural frequency is approximately 28,000 Hz (1,680,000 CPM) and the maximum useful frequency is about 10,000 Hz (600,000 CPM). Stud mounting always provides the best possible frequency response characteristics for either an accelerometer or velocity pickup.

Moving down to Type 4 with the accelerometer on an adhesive mounting, notice the difference when a good material such as Cyanoacrylate cement was used as opposed to when a soft glue with too great a thickness was employed. In the case of the Cyanoacrylate cement, it gave almost as good a frequency response as that obtained with Type 1 ending up with a mounting natural frequency of 26,000 Hz (1,560,000 CPM) and a flat response out to approximately 9000 Hz (540,000 CPM). However, the soft glue provided a flat response out to only 6000 Hz (360,000 CPM).

The spectra for the Type 5 mounting point out an even greater difference if too thick a double sided adhesive tape is used. In one case, the thin double sided tape provide a 19,000 Hz natural frequency and 5000 Hz useful frequency (300,000 CPM), whereas the flat response drops all the way down to only 500 Hz (30,000 CPM) when too thick a double sided adhesive tape is employed.

Mounting Types 6 and 7 should be carefully reviewed by all those taking periodic vibration surveys on key machines. A magnet is used under the accelerometer in Type 6 whereas a probe is attached directly to the accelerometer in the case of Type 7. Both mountings result in a drop in mounting natural frequency and in flat response. However, the dropoff for the hand-held probe is considerably worse, particularly for those wanting to evaluate the condition of rolling element bearings.

In the case of the frequency response for the magnet mount in Type 6, this response is fairly typical of that for most of the accelerometer/ magnet systems offered by data collector vendors today. Its flat response is out to about 2000 Hz (120,000 CPM), but it can be successfully used out to as high as 3000 Hz (180,000 CPM) if the analyst takes careful precautions when mounting his magnet (making sure it is securely fastened to the machine with no rocking motion). Remember that a rocking motion of only .001 inch at 1800 RPM will produce an equivalent "false velocity" of almost .10 in/sec; and only .006 inch rocking at 1800 CPM will produce nearly .60 in/ sec! Thus, magnet mounted accelerometers can produce good, repeatable measurements if they are mounted securely.

On the other hand, the frequency response for the Type 7 probe mount shown in Figure 6.09HH is considerably lower than either the Type 1 stud mount (or the Type 6 magnet mount for that matter). Note that the mounting natural frequency drops from 28,000 Hz (1,680,000 CPM) in the case of Type 1 down to only 1650 Hz (99,000 CPM) for this hand-held probe. Even more dramatic, its flat response drops from 10,000 Hz (600,000 CPM) for Type 1 down to only 450 Hz (27,000 CPM) when the probe is attached to the accelerometer. Another concern is seen by looking at what happens in Type 7 when actual forcing frequencies are present higher than the 1650 Hz natural frequency. Note that the response "drops off the table". That is, if there are

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frequencies related to bearing health beyond about 2500 Hz (150,000 CPM), the hand-held probe may not even detect their presence.

Finally, one of the other problems with hand-held probes in general is the fact that they will display vibration frequencies that appear to be of real concern right in the middle of the zone where so many bearing component natural frequencies and defect frequencies will occur between approximately 30,000 and 120,000 CPM. The reason for this is the great resonant amplification caused by the probe itself when any of these frequencies are present in this wide resonant amplification zone. The Type 7 spectrum shows that it can amplify frequency levels by as much as 26 dB (or a factor of 20 times higher). Therefore, this can lead to continually deceiving the analyst into believing bearing problems are present on one machine after another when, in fact, there may actually be little or nothing wrong with most all of the bearings.

Therefore, when any measurements are desired requiring a maximum frequency (F_{MAX}) greater than 30,000 CPM, do not use a hand-held probe (unless there is no other way the bearing housing can safely be reached. If measurements up to 240,000 CPM are required, use at least a magnet mount. And, if measurements higher than 300,000 CPM are desired, a stud-mount (or at least a "Quick-Connect Mount") will have to be used to obtain meaningful, repeatable data, and to detect potentially significant problems which might be indicated at frequencies above approximately 1500 - 2000 Hz (90,000 - 120,000 CPM) which include not only bearing frequencies, but also gear mesh frequencies (and harmonics), rotor bar pass frequencies (and harmonics), lubrication-related frequencies, synchronous motor electrical problems at the coil passing frequency (and harmonics), dry whip in a plain bearing, etc.

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FIGURE 6.09HH
EFFECT ON USEFUL FREQUENCY LIMIT ON MOUNTING OF ACCELEROMETER

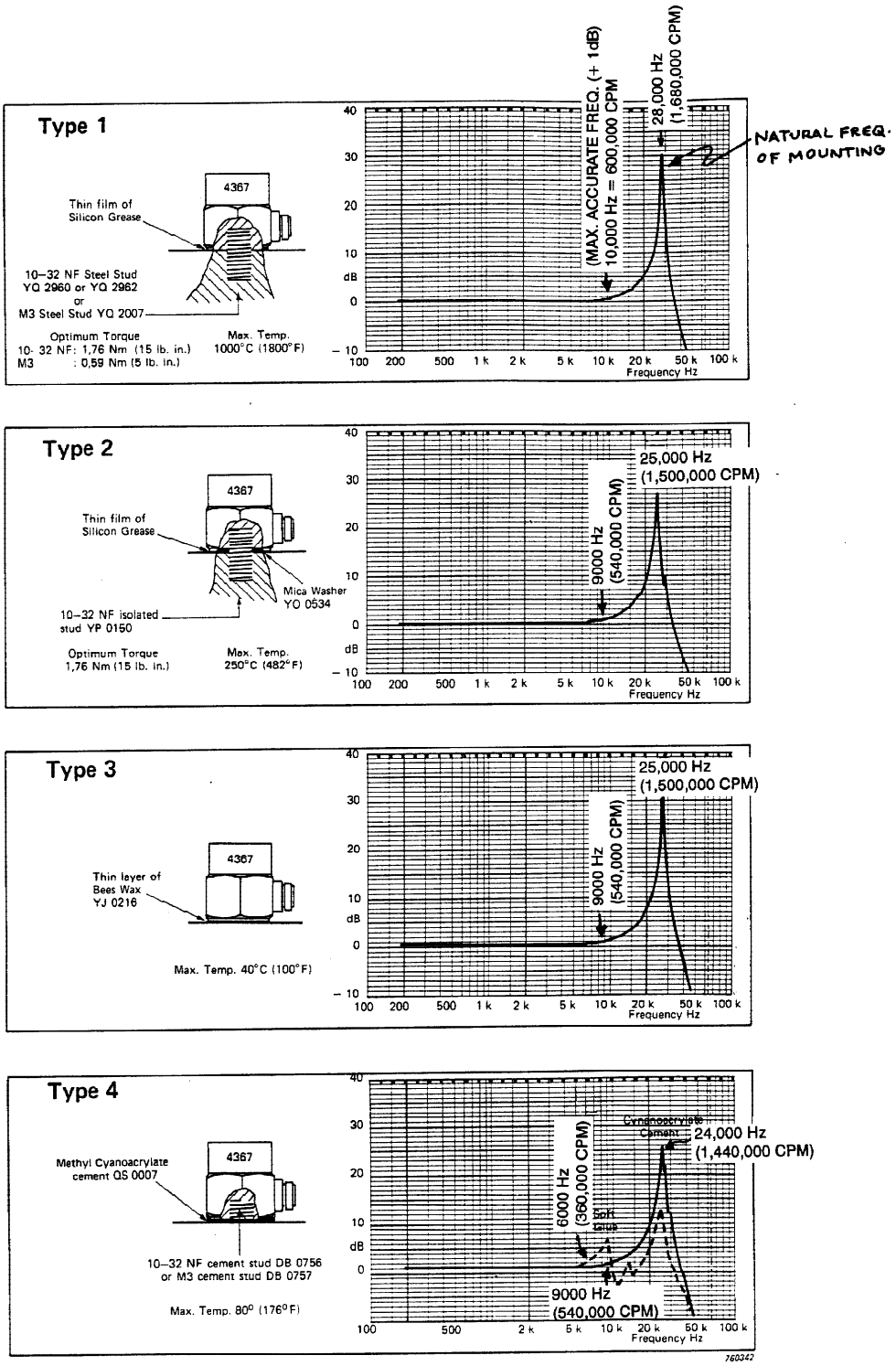
Type 1
4367
Thin film of Silicon Grease
10–32 NF Steel Stud YQ 2960 or YQ 2962 or M3 Steel Stud YQ 2007
Optimum Torque 10- 32 NF: 1,76 Nm (15 lb. in.) M3 : 0,59 Nm (5 lb. in.)
Max. Temp. 1000°C (1800°F)
(MAX. ACCURATE FREQ. (+ 1dB) 10,000 Hz = 600,000 CPM
28,000 Hz (1,680,000 CPM)
NATURAL FREQ. OF MOUNTING
Type 2
4367
Thin film of Silicon Grease
Mica Washer YO 0534
10–32 NF isolated stud YP 0150
Optimum Torque 1,76 Nm (15 lb. in.)
Max. Temp. 250°C (482°F)
9000 Hz (540,000 CPM)
25,000 Hz (1,500,000 CPM)
Type 3
4367
Thin layer of Bees Wax YJ 0216
Max. Temp. 40°C (100°F)
9000 Hz (540,000 CPM)
25,000 Hz (1,500,000 CPM)
Type 4
4367
Methyl Cyanoacrylate cement QS 0007
10–32 NF cement stud DB 0756 or M3 cement stud DB 0757
Max. Temp. 80° (176°F)
Cyanoacrylate Cement
6000 Hz (360,000 CPM)
Soft tetus
24,000 Hz (1,440,000 CPM)
9000 Hz (540,000 CPM)
40
30
20
10
dB
0
– 10
100 200 500 1 k 2 k 5 k 10 k 20 k 50 k 100 k
Frequency Hz
760342

Fig. 4.2. Methods of mounting B & K piezoelectric accelerometers and examples of typical frequency responses obtained

(Reference: Piezoelectric Accelerometer & Vibration Preamplifier Handbook; Bruel & Kjaer; Revision March 1978; pages 52-53.)

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FIGURE 6.09HH
(Continued)

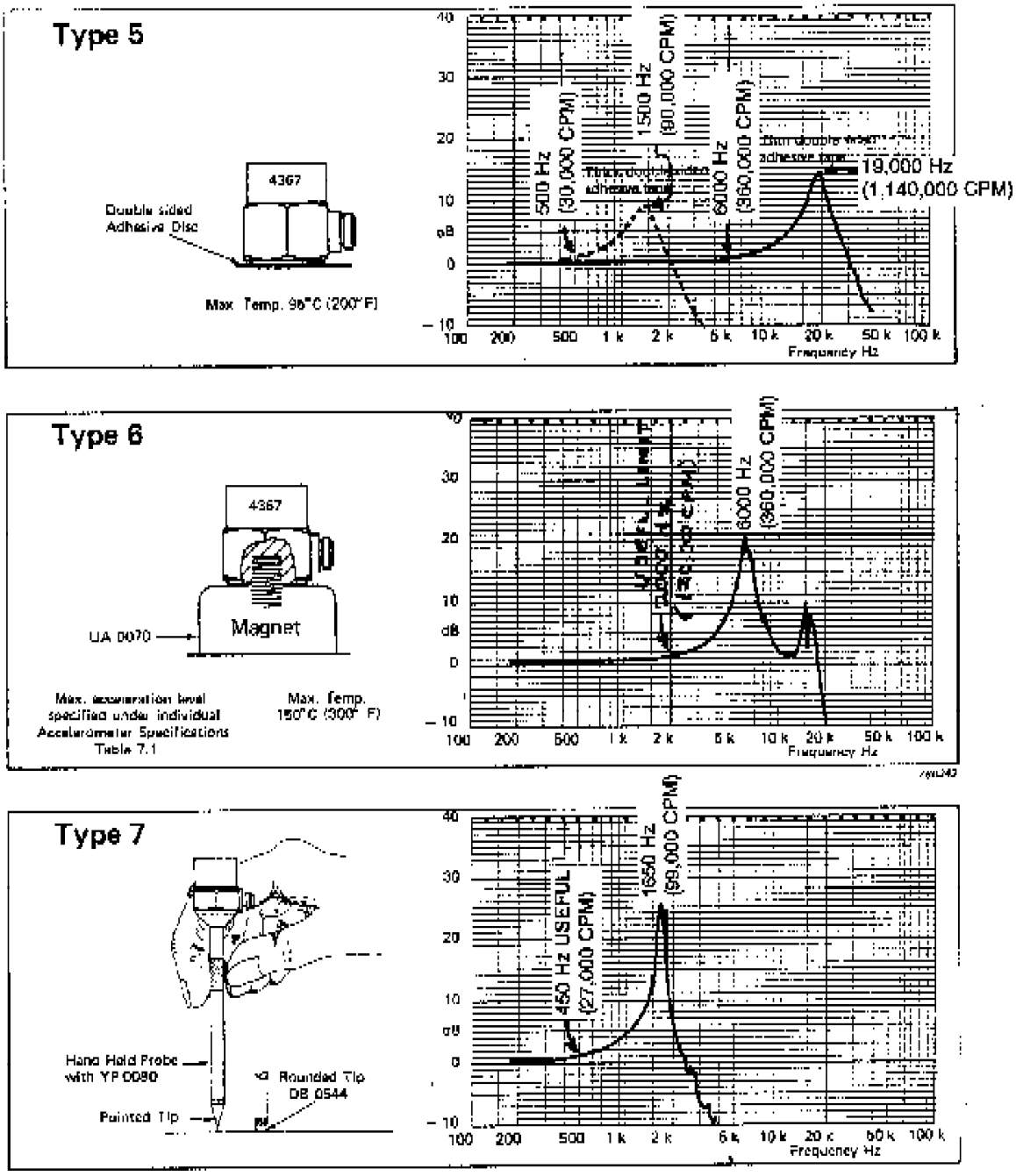

Fig. 4.3. Methods of mounting B & K piezoelectric accelerometers and examples of typical frequency responses obtained

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6.095 RECOMMENDATIONS ON WHEN ROLLING ELEMENT BEARINGS SHOULD BE REPLACED:

With a knowledge now of the various paths which can be taken by rolling element bearings to failure and how these paths affect the spectral content, this can go far in answering the age-old question of "When do we need to replace the bearings?" First, the analyst must have several questions answered for him such as:

1. How critical is this machine?

2. What is the cost per hour for it to be down?

3. What are the costs of replacement bearings (and how long will it require for the replacement bearing to arrive once an order has been placed)?

4. Is bearing deterioration affecting the machine productivity/quality performance?

5. If we decide not to replace it now, what would be the consequences were it to fail catastrophically?

6. What exactly is causing the bearing to fail (unbalance, misalignment, lubrication problems, excessive temperature, etc.)?

7. Is there any backup for this machine should it fail before predicted?

8. What is the production work load now? Is there a letup in the near future?

9. Will there be a scheduled downtime soon (if so, will this bearing "survive" until then)?

10. Am I positive this bearing has a problem?

These are just some of the questions that must be answered before a decision can be made. Again, note that the analyst himself should not attempt to answer all these questions himself.

With these questions and answers in mind, analysis on when to replace the bearings should likely be broken down into - (a) general machines of moderate criticality having reasonably priced replacement components with nominal lead times; or (b) large, highly critical machines, with expensive replacement components with long lead times.

Since this paper has shown there are a number of different failure scenarios through which rolling element bearings can pass, each of these must be considered separately for both the Type A (Noncritical, General Machinery) and Type B (Critical, Expensive Machinery). Table 6.09B on the following page has taken each of these into account and is offered to guide the analyst in his decision of whether he must take action and when he should do so.

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TABLE 6.09B
WHEN TO REPLACE ROLLING ELEMENT BEARINGS ON NONCRITICAL GENERAL MACHINERY VERSUS CRITICAL, EXPENSIVE MACHINERY

FAILURE SCENARIO	TYPE A MACHINES (NONCRITICAL, GENERAL MACHINERY WITH NOMINAL LEAD TIMES)	TYPE B MACHINES (CRITICAL, EXPENSIVE MACHINERY WITH LONG LEAD TIMES)
A. Primary Failure Stage Scenario (Most Rolling Element Bearings)	(A-1) If any defect frequency is present ≥30% OA Alarm OR (A-2) If 3 or more defect frequencies are present each of which are ≥ 15% OA Alarm OR (A-3) If 1X sidebands ≥ 10% OA Alarm are present around any defect frequency which itself ≥ 20% OA Alarm	(B-1) If any defect frequency is present ≥ 20% OA Alarm OR (B-2) If 2 or more defect frequencies are present each of which are ≥ 10% OA Alarm OR (B-3) If 1X sidebands ≥ 7.5% OA Alarm are present around any defect frequency which itself ≥ 15% OA Alarm
B. One Pronounced Fault Exciting Dominant Defect Frequency (Metallurgical Problem)	Rule (A-1) OR Rule (A-2) OR (A-4) If 1X sidebands ≥ 10% OA Alarm are present around any defect frequency which itself ≥ 25% OA Alarm (Since only 1 dominant defect frequency exists)	Rule (B-1) OR Rule (B-2) OR (B-4) If 1X sidebands ≥ 7.5% OA Alarm ips are present around any defect freq. which itself ≥ 15% OA Alarm (Since only 1 dominant defect frequency exists)
C. Continual Wear Throughout Periphery of Raceway (No Sidebands Developed) (This Should Not Be Present At All)	Rule (A-1) OR Rule (A-2) OR Rule (A-3)	Rule (B-1) OR Rule (B-2) OR Rule (B-3)
D. When Defect Frequency Appears As Sideband Frequency Itself (Roller/Ball Or Cage Problem)	(A-5) When any difference frequency ≥ 30% OA Alarm equal any defect frequency OR (A-6) If 3 or more difference frequencies ≥ 15% OA Alarm equal any defect frequency OR Rule (A-3)	(B-5) When any difference frequency ≥ 20% OA Alarm equal any defect frequency OR (B-6) If 2 or more difference freq. ≥ 10% OA Alarm equals any defect frequency OR Rule (B-3)
E. 1X RPM Multi-harmonics Ending With Severe Mechanical Looseness or Bearing Turning on Shaft (Undersized Shaft)	(A-7) If any 5 running speed harmonics exists each of which are ≥ 15% OA Alarm	(B-7) If any 4 running speed harmonics exists each of which are ≥ 10% OA Alarm
F. Excessive 1X RPM Sidebanding About Race Frequencies Ending Up With 1X RPM Multi-harmonics (This Should Not Be Present At All)	Rule (A-1) OR Rule (A-2) OR Rule (A-3) OR Rule (A-7)	Rule (B-1) OR Rule (B-2) OR Rule (B-3) OR Rule (B-7)

NOTES:
1. "Defect Frequencies" = Refers to any defect frequency whether it is a fundamental defect frequency or a harmonic of a defect frequency.
2. "Difference Frequency" = The mathematical difference between any 2 frequencies in the spectrum.
3. "OA Acceptance =Acceptance Level of Overall Velocity (in/sec) specified for particular machine type (rms or peak).
4. Table X.X is offered as a guide to help the analyst decide when bearing replacement is required. It can be adapted by analysts within a specific plant (as required) to further enhance this important decision which will affect not only Maintenance, but also Production.

6.10 FLOW-INDUCED VIBRATION

Unlike many of the other mechanical vibration problems such as unbalance, misalignment, resonance and so forth, flow-induced vibration is often very dependent upon operating conditions. That is, it can be very affected by load depending on the type of machine and the work being performed by the machine. Flow-induced vibration includes each of the following:

- Hydraulic and Aerodynamic Forces
- Cavitation and Starvation
- Recirculation
- Flow Turbulence
- Surge
- Choking

Each of these problems will be discussed in Section 6.101 through 6.106. Each of these phenomena are capable of generating extremely high energy levels and must be avoided by proper design and operation of the machines involved. In addition, each of them generate particular frequencies of vibration which again are often dependent upon the load and other operating conditions. Following below will be a discussion on each type of flow-induced vibration.

6.101 Hydraulic and Aerodynamic Forces:

Various types of machines including pumps, blowers, turbines, vacuum pumps and so forth inherently produce hydraulic or aerodynamic forces as their impellers impart work on the fluid (liquid or gaseous) they are handling. In most cases, the vibration generated from hydraulic and aerodynamic forces is not a problem unless it happens to excite resonant frequencies, if a potentially destructive phenomenon known as cavitation occurs, or if vibration generated by the impellers themselves becomes excessive subjecting expensive machine components to undesirably high vibration. Also, when vibration at blade pass frequency suddenly jumps, it can indicate something has gone wrong with the driven part of the machine. This will be covered below.

In general, hydraulic and aerodynamic forces are generated as per the following equation:

Blade (or Vane) Pass Frequency = | BPF = #Vanes X Impeller RPM (CPM or Hz) |

These forces are generated by a pressure variation or pulse each time a blade (or vane) loads or unloads as it passes nearby stationary components (such as diffuser vanes or discharge volutes). That is, if one were at a fixed point in space and he observed a six vane pump impeller, he would "feel" each of 6 pulses as the impeller made one revolution (in this case, 6 events per revolution).

Hydraulic and aerodynamic force vibration exhibits each of the following characteristics:

1. Significant vibration both at blade pass frequency and blade pass harmonics will be generated if an impeller is not centrally located within its housing and properly aligned with diffusers. For example, if the gap between impeller and diffuser vanes is .25" on one side, but .50" on the other a whole series of blade pass frequencies can be generated, likely at high vibration levels. Clearances all throughout the periphery should be identical unless there are specific design reasons to depart from this (as is the case with some types of pumps and fans).

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2. It is important to ensure that neither the blade pass frequency nor blade pass harmonics coincide with natural frequencies of the rotor or support structure. Excessive vibration can result if allowed to do so.

3. A great increase in blade pass frequency will occur if welds securing diffuser vanes experience failure allowing the diffusers to shift position slightly relative to the impeller. This will affect hydraulic/aerodynamic flow which should be obvious from a waterfall spectrum trending vibration spectra from one survey to the next. Such an example is given in Figure 6.10A. Note from the lower plot that Blade Pass vibration at 14,310 CPM (16X RPM) significantly increased 10X higher from .010 to .100 in/sec between one survey and the next; then almost doubled again up to .193 in/sec by the next survey due to diffuser weld fatigue.

4. Similarly, a great increase in blade pass vibration will be created if a pump impeller wear ring seizes to the shaft.

5. Blade pass frequency in some particular fan types is particularly sensitive to damper settings. In other words, with these fans, just altering the damper setting alone can greatly affect blade pass vibration. Here again, this vibration alone is not particularly destructive to any machine component unless it should exceed approximately .30 to .40 in/sec peak velocity for most blower types.

6. Another frequency which can sometimes be generated on centrifugal machinery outfitted with rotating vanes and stationary diffusers is known as the Blade Rate Frequency (BRF). The blade rate frequency has to do with the number of times and rate at which rotating and stationary vanes coincide with one another. When they do coincide, there can be a pronounced pulse due to compression of the fluid (gas or liquid) between the rotating and stationary vanes. These pulses from simultaneous coincidence of two or more sets of vanes can be much stronger than if only one rotating and stationary vane were directly in line with one another at a given instant of time. This is why it is not a good idea to have numbers of rotating vanes and diffusers which have common denominators of one another.

   The Blade Rate Frequency (BRF) is as follows:

$$BRF = \frac{(\#\text{Impeller Vanes})(\#\text{Diffuser Vanes})(RPM)}{K}$$

   where:

   BRF = Blade Rate Frequency (CPM or Hz)
   K = Highest Common Factor of #Impeller Vanes and #Diffuser Vanes

For example, if a machine had 18 impeller vanes and 24 diffuser vanes, K would equal to 6 so that:

$$BRF = \frac{(18\ \text{Impeller Vanes})(24\ \text{Diffuser Vanes})(RPM)}{6}$$

$$= 72 \times RPM$$

Thus, BRF (72 X RPM) would be 4X higher than BPF (18 X RPM) in this case. And, as was pointed out above, this machine would likely suffer much higher pulsations due to the fact that more than one set of impeller and diffuser vanes would line up with one another (in this case, 6 impeller vanes would simultaneously be directly opposite diffuser vanes at angles of 0°, 60°, 120°, 180°, 240° and 300°), resulting in pronounced pulsations at Blade Rate Frequency (BRF). If instead there were either 17 impeller vanes or 25 diffuser vanes, at no instant in time would more than one set of impeller and diffuser vanes line up with one another, and therefore, the machine would likely experience lower vibration.

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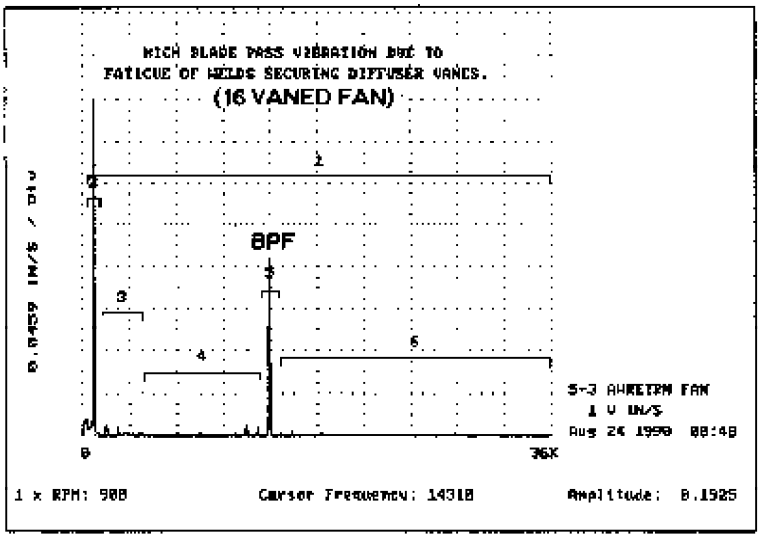

001% x 0.500 = 0.0050 1 x RPM: 900.00

Machine	Pos	Dir	Units	Date
5-3 AWRETRN FAN	1	V	IN/S	Aug 24 1990 08:48

Frequency	1 x RPM	Amplitude
270	0.30	0.0175
630	0.70	0.0150
900	1.00	0.3675
1800	2.00	0.0125
2700	3.00	0.0075
3600	4.00	0.0050
12510	13.90	0.0125
13410	14.90	0.0075
14310	15.90	0.1925
16110	17.90	0.0050

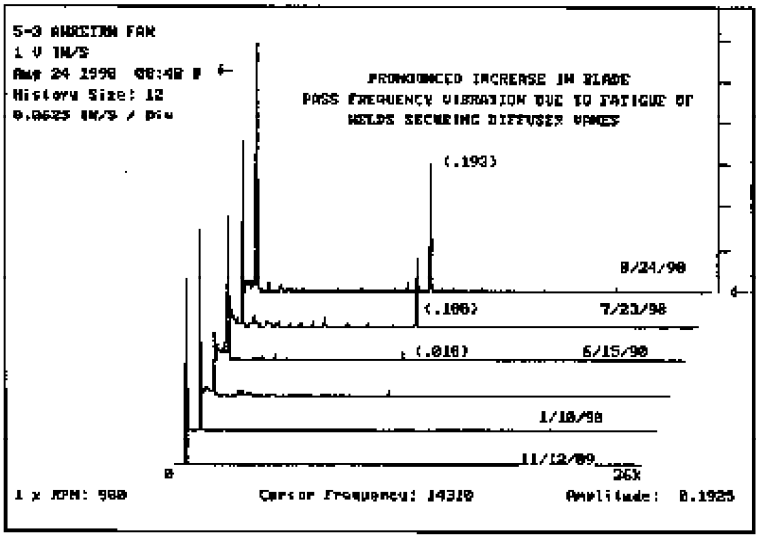

FIGURE 6.10A
HIGH BLADE PASS VIBRATION DUE TO FATIGUE OF WELDS SECURING DIFFUSER VANES (16 VANED FAN)

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6.102 Cavitation and Starvation:

Cavitation is a common problem with centrifugal pumps and can be quite destructive to internal pump components. Often, pumps subjected to continuing cavitation will have badly pitted and/ or eroded impellers. In some cases, cavitation can completely erode away impeller vanes.

Cavitation most often occurs when a pump is operating with excess capacity or low suction pressure. In essence, since the pump is "starved", the fluid entering the pump is almost literally "pulled apart" as it attempts to fill the voids which exists. This creates pockets of nearly perfect vacuum which are prone to collapse or implode quickly. As they do so, these implosions cause impacts which excite local natural frequencies of the impeller and nearby components. Since these implosions may occur at random intervals all throughout the pump or connected piping, the vibration will be highly random both in amplitude and in frequency.

Cavitation exhibits each of the following characteristics:

1. Classic spectra indicating cavitation have random, broad band energy most often between approximately 20,000 CPM up through approximately 120,000 CPM. It can appear as "white noise" with no discernible frequency content, or at times with vane pass frequencies superimposed. That is, several blade pass frequency harmonics sometimes will appear along with the random, higher frequency vibration.

2. When cavitation occurs it will increase ultrasonic measurements (spike energy, HFD, shock pulse, etc.). Figure 6.10B shows a spectrum indicative of cavitation problems. Note not only the random broad band vibration, but also the high levels of spike energy on both pump bearings (positions 3H and 4H which were both in alarm on spike energy). Normally, cavitation will induce high spike energies on both pump bearings, whereas lubrication or wear problems of one bearing usually will show high spike energy only on the one bearing with the problem.

3. Cavitation will most often generate unique noise. Milder cavitation normally sounds like sand being pumped whereas more severe cavitation will actually sound as if gravel is being passed through the pump (therefore, if the analyst hears this type of sound during his vibration survey when evaluating a pump, he should make a note of this during his route and/or enter an inspection code indicating this observation if his data collector has this capability; then, when back at the office, if the analyst sees both the random, high frequency vibration and notes the abnormal noise, it will greatly improve the validity of his diagnosis).

4. Starvation is the aerodynamic counterpart to cavitation. Like cavitation, it involves insufficient air flow relative to fan capacity. It normally involves damper settings and at times, improper application of equipment. Typical starvation spectra are almost identical to those of cavitation (higher frequency, broad band energy) which sometimes can be excessive causing great vibration in suction and discharge ducts in addition to the fan itself.

6.103 Recirculation:

In the case of a pump, recirculation is just the opposite of cavitation. Recirculation can occur when a pump is operated at too low a capacity or too high suction pressure. When it occurs, a portion of the fluid returns from the discharge to the impeller as the pump attempts to move an excessive amount of fluid through the pump. This brings about a reverse flow and results in 2 or more fluids moving in opposite directions causing noise and vibration. Recirculation exhibits the following characteristics:

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1. Recirculation generates vibration spectra very similar to those caused by cavitation. That is, it will often have random, higher frequency vibration, sometimes superimposed with vane pass frequencies.

2. Unlike cavitation, recirculation will not likely cause wear or erosion of pump components. However, if the vibration becomes excessive, it can damage bearings, wear rings, valves and other related components.

6.104 Flow Turbulence:

Flow turbulence occurs when something interferes with or resists the normal flow of liquids or gases through pumps, fans, compressors or vacuum pumps. Such flow disturbances might be caused by obstructions in the duct or pipe, sharp right angle turns, abrupt changes in diameter, etc. Correction of these problems would include removal of obstructions, insertion of turning vanes if a duct presents too sharp a right angle turn, and provision of a longer, stepped change in diameter where necessary to reduce duct or pipe cross section.

Figure 6.10C shows a typical spectrum indicating flow turbulence. Flow turbulence problems exhibit the following characteristics:

1. Although flow turbulence can generate random, higher frequency vibration, it normally will cause random, lower frequency vibration below 1X RPM. In the case of fans and blowers, reference 8 reports that flow turbulence will create random, low frequency vibration concentrated in a range from approximately 50 CPM up to 2000 CPM which will represent the natural frequencies of the machine, structure and duct work. In the case of pumps, Reference 9 reports that frequencies ranging from .55X RPM up to .78X RPM will be excited in the case of hydraulic instability, problems with wear rings, seal trouble, or difficulties with the balancing disc or drum.

2. Amplitudes and frequencies of flow turbulent vibration will be erratic, sometimes pulsating widely.

3. Sometimes noise produced by flow turbulence will be very high although the vibration accompanying it will be relatively low on the machine itself. The reason for this is that the turbulence condition itself exists outside the machine.

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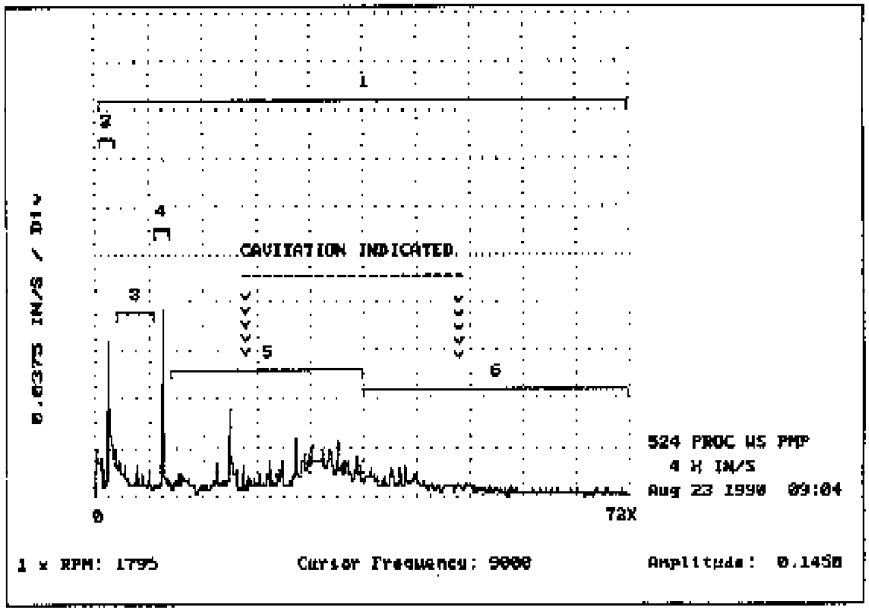

005% x 0.500 = 0.0250 1 x RPM: 1795.00

Machine	Pos	Dir	Units	Date
524 PROC WS PMP	4	H	IN/S	Aug 23 1990 09:04

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
180	0.10	0.0375	32580	18.15	0.0425
900	0.50	0.0300	33120	18.45	0.0250
1800	1.00	0.1200	33480	18.65	0.0275
2700	1.50	0.0375	34020	18.95	0.0275
3240	1.81	0.0275	35100	19.55	0.0300
9000	5.01	0.1450			
16200	9.03	0.0250			
18000	10.03	0.0675			
23220	12.94	0.0275			
25020	13.94	0.0275			
26820	14.94	0.0450			
27720	15.44	0.0275			
28080	15.64	0.0325			
28620	15.94	0.0325			
29160	16.25	0.0400			
30600	17.05	0.0350			
31500	17.55	0.0350			

T - Threshold adjust Q - Quit

Machine Report (Latest Data) for: 524 PROC WS PMP

Date	Time	Pos	Dir	Ampl	Units	Change	Alarm	Inspection
Aug-23-90	09:03	1	A	.048	IN/S	+33%	.300	
Aug-23-90	09:03	1	H	.096	IN/S	-20%	.300	
Aug-23-90	09:03	1	H	.010	g/SE	+16%	.600	
Aug-23-90	09:03	1	O	.067	IN/S	-----%	.300	
Aug-23-90	09:03	1	V	.054	IN/S	-2%	.300	
Aug-23-90	09:03	2	A	.101	IN/S	80%	.300	
Aug-23-90	09:03	2	H	.123	IN/S	15%	.300	
Aug-23-90	09:03	2	H	.037	g/SE	23%	.600	
Aug-23-90	09:03	2	O	.125	IN/S	-----%	.300	
Aug-23-90	09:04	2	V	.063	IN/S	9%	.300	
Aug-23-90	09:04	3	A	.250	IN/S	18%	.300	
Aug-23-90	09:04	3	H	.212	IN/S	-21%	.300	
Aug-23-90	09:04	4	H	+.640	g/SE	21%	.600	PUMP INBOARD (3H) & OUTBOARD (4H) BEARINGS
Aug-23-90	09:04	3	V	.195	IN/S	17%	.300	
Aug-23-90	09:04	4	A	+.307	IN/S	76%	.100	
Aug-23-90	09:04	4	H	+.348	IN/S	7%	.300	
Aug-23-90	09:04	4	H	+0.74	g/SE	4%	.600	
Aug-23-90	09:04	4	V	.388	IN/S	27%	.100	

18 lines this report

FIGURE 6.10B
DEVELOPMENT OF CAVITATION IN A WATER SUPPLY PUMP

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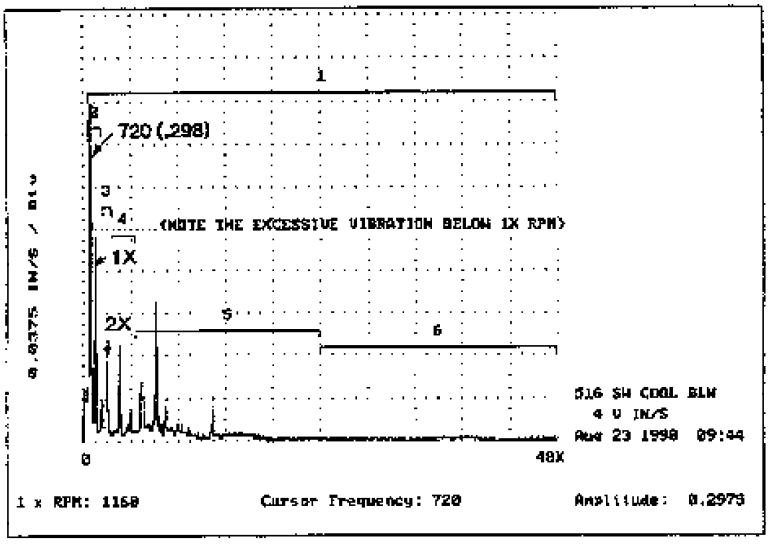

003% x 0.500 = 0.0150 1 x RPM: 1160.00

Machine	Pos	Dir	Units	Date
516 SW COOL BLW	4	V	IN/S	Aug 23 1990 09:44

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
120	0.10	0.0475			
720	0.62	0.2975			
1200	1.03	0.1800			
1800	1.55	0.0375			
2400	2.07	0.0700			
3720	3.21	0.0850			
4800	4.14	0.0275			
5880	5.07	0.0525			
6120	5.28	0.0400			
6600	5.69	0.0150			
7320	6.31	0.1225			
7680	6.62	0.0200			
7920	6.83	0.0150			
8280	7.14	0.0300			
9600	8.28	0.0150			
9960	8.59	0.0150			
13080	11.28	0.0400			

FIGURE 6.10C
PROBABLE TURBULENT FLOW PROBLEM INDICATED ON THIS BLOWER

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6.105 Surge:

One of the more important problems plaguing high speed centrifugal and axial-flow compressors is surge. Typically a compressor experiencing surge is operating outside design limits. Surge occurs when the discharge pressure is too high or if the volumetric mass flow rate is too low relative to design conditions for a particular operating speed. When this occurs, the gas flow will actually reverse directions in the compressors. When the surge is only minor, this flow reversal may occur only at the boundary layers of the impeller blades. However, when full surge develops, the entire fluid flow will reverse directions and will flow from the discharge to the inlet. Surge must be prevented in a compressor due to the fact that it can wreak extensive damage. Fortunately, many electronic controls are available today which can make almost instantaneous adjustments within the machinery to prevent surge. Surge exhibits the following characteristics:

1. Minor surge causes a noticeable increase at blade pass frequency and harmonics within only a time period of seconds. Such blade pass frequencies may double or even triple in amplitude.

2. Fully developed surge will most often lift up the entire spectrum. That is, the entire "noise floor" of the spectrum will be lifted, creating high amplitude, random response covering a broad frequency range. This is caused by turbulent flow within the compressor exciting a series of natural frequencies including those of the impeller wheels, rotor blades, diffuser blades, casing, shaft, gearing and other components. Left uncorrected, surge can wreck a compressor within a short period of time.

6.106 Choking:

Choking (sometimes called "stone walling") is essentially the opposite of surging in a compressor. Choking occurs when discharge pressures are too low creating high velocities in the diffuser section. When such flow velocities approach Mach 1, a turbulent flow will occur between the diffuser blades which will have the effect of blocking the fluid flow. When this occurs, not only will there be an increase in vibration due to the turbulent flow, but also there will a significant drop in compressor efficiency and pressure ratio. Vibration spectral characteristics of choking will be essentially identical to those of surging. Therefore, one will have to check other operating parameters such as pressure, mass flow, and so forth in order to differentiate which problem is occurring.

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6.11 GEAR PROBLEMS

A variety of gear problems can be detected by vibration analysis. These include each of the following:

- Gear Tooth Wear
- Excessive Tooth Load
- Gear Eccentricity and/or Backlash
- Cracked, Chipped or Broken Gear Teeth
- Hunting Tooth Problems

Each of these problems will be discussed in Sections 6.111 through 6.116. First, several remarks in general should be made about vibration diagnostics on gears.

One of the key frequencies of interest when evaluating gear health will be gear mesh frequency (#teeth X RPM). However, it is important to point out gear mesh frequency (GMF) is not a fault or defect frequency as is the case with bearing defect frequencies. All meshing gears generate gear mesh frequencies of some amplitude or another. In addition, all gear mesh frequencies will have sidebands of some amplitude spaced at the RPM of each mating gear in the mesh. However, if the gears are in good health and are properly aligned with one another (insignificant misalignment, backlash or gear eccentricity), amplitudes of GMF and its harmonics along with those of sidebands should be low, particularly those of the sidebands.

Figure 6.11A shows a standard setup for analyzing a right angle, double reduction gearbox outfitted in this case with bevel and helical gears. The example illustrated in Figure 6.11A will be used to make several comments:

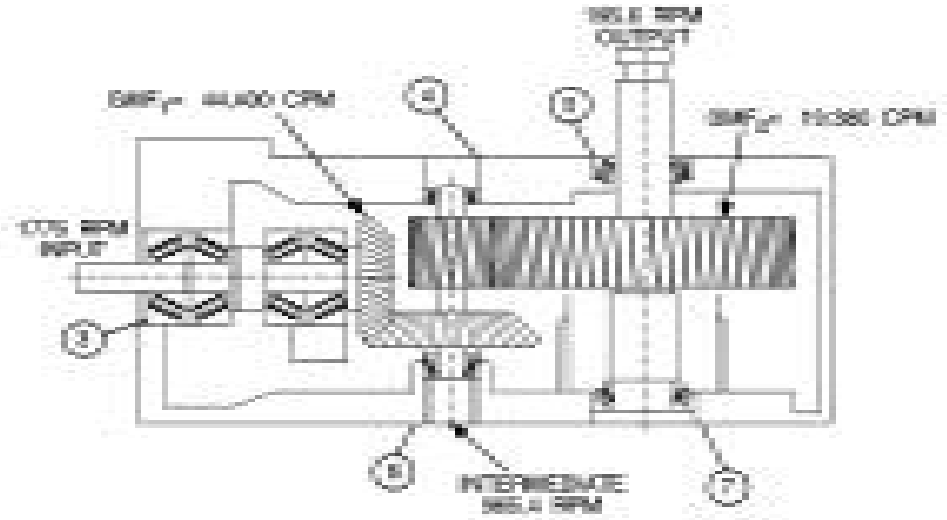

FIGURE 6.11A
STANDARD SETUP FOR ANALYSIS OF A RIGHT ANGLE DOUBLE REDUCTION GEARBOX

1. Vibration measurements should be made on each bearing housing which is accessible. Figure 6.11A shows measurements being taken on the double row bearing at position 3 as well as on each of the four tapered roller bearings in positions 4 through 7. The important point is that the transducer should be placed as near as possible to the bearings supporting the gears themselves. At times, this may involve measurements at a distance from the bearings. However, in this case, ensure that a frame or internal web that goes directly to the bearing housing is located and measurements made as close to these as possible.

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2. Measurements should be taken in all 3 orthogonal directions (horizontal, vertical and axial), particularly since some gears generate forces predominately in one direction or the other which may not be the same from one survey to the next, depending on load.

3. In general, helical, herringbone and bevel gears generate significant vibration in the axial direction. Often, the best condition measurement on these gear types is in this axial direction.

4. In general, spur gears are best evaluated in radial directions, but can sometimes have significant axial vibration as well, particularly if there is a problem with tooth alignment. At times, very high frequency measurements will have to be made to evaluate gear condition. One should always evaluate frequencies at least up to 3X gear mesh frequency (for regular PMP surveys, use an F_{MAX} = 3.25X GMF). If the number of teeth are unknown, set F_{MAX} = 200X RPM using the speed of the shaft at each particular bearing positions being measured.

   The author has often been involved on machines where little vibration was occurring at the fundamental gear mesh frequency (GMF), but where much higher vibration on the order of 10X those at the fundamental were occurring at either 2X gear mesh or 3X gear mesh frequency (2GMF or 3GMF). Therefore, potentially significant problems would have been overlooked had measurements not been made up in these frequency regions.

5. The analyst may sometimes have to employ more than one accelerometer on a gear box due to potentially high frequencies that might occur on one or more of the meshes. For example, if the fundamental gear mesh frequency was on the order of 1,200,000 CPM (20,000 Hz), he would evaluate balance, alignment, looseness, etc. Then, he would have to make a whole set separate measurements evaluating the gears with special accelerometers having much higher frequency capability. Of course, the converse could also be said if one were evaluating the output shaft which was at very low RPM (below 100 RPM) with a moderately high gear mesh frequency. In this case, one might have to employ a special, low frequency seismic accelerometer for the lower frequency problems and the general purpose accelerometer for the gear mesh measurements.

6. In most PMP programs using computer software, each of 2 measurements having 2 different frequency ranges must be made at each gear location because of the widely varying frequency between the lower speed harmonics and the gear mesh frequencies themselves. In these cases, he would use a lower F_{MAX} to evaluate such problems as unbalance, misalignment, looseness, electrical, etc. and a completely different set of measurements to evaluate gear health.

7. For a given mesh of gears having one common gear, the gear mesh frequency will always be the same no matter whether 2, or up to 5 or 6 gears happen to be in a common mesh. For example, in many of today's centrifugal air compressors, there is one bull gear which meshes with each of 4 pinions that are mounted on the first through fourth stage impellers. In this case of 5 meshing gears, there is only one gear mesh frequency.

8. Of course, the gear mesh frequency is different for each different mesh of gears (see Figure 6.11A). For example, one shaft may have one gear in a mesh on one end and another gear on its opposite end meshing with other gears. Each of these meshes will have individual gear mesh frequencies.

9. Referring to Figure 6.11A, an analyst should always make a drawing such as this showing each of the positions, the RPM of each shaft and the gear mesh frequency at each mesh. This will go far in helping him with his analysis.

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10. A multi-stage gear box should actually be treated as several individual vibration problems, each with its own unique set of operating speeds and gear mesh frequencies. For example, using Figure 6.11A, one problem should be analyzed on the first mesh of 44,400 CPM with the input and output shaft speeds being 1775 RPM and 965.4 RPM, respectively. In all likelihood, measurements on this input shaft will likewise contain vibration at the lower speed gear mesh and the output shaft speed. However, in most cases, he should neglect vibration being transmitted from the other meshes with the exception of resonances on this member being excited by vibration transmitted from the others.

Following below will be separate discussions on each of the gear problems previously listed:

6.111 Gear Tooth Wear:

Figure 6.11B shows a spectrum indicating wear of gear teeth. In this case, wear does not refer to a chipped, broken or cracked tooth. Instead, it refers to surface damage across the tooth face. Worn gear teeth exhibit the following characteristics:

1. The key indicator of gear tooth wear is not the gear mesh frequency, but instead the gear natural frequency. In reality, there is of course more than one gear natural frequency including separate ones for the driver and driven gears as well as a set of those when the gears are meshing with one another. These natural frequencies can be identified by performance of impulse natural frequency tests outlined in Section 6.05. Like everything else in nature, when a member is impacted, it will respond at its natural frequency. In the case of gears, their natural frequencies respond each time a defective tooth hits or impacts as it goes into and out of mesh. The key here is that these natural frequencies will be modulated by the impact repetition rate which will correspond to the speed of the worn gear.

2. When significant tooth wear occurs, not only will sidebands appear about the gear mesh frequencies, but also about the gear natural frequencies. In the case of those around the gear mesh frequencies, the amplitude of the sidebands themselves is a better wear indicator than the amplitude of GMF frequencies.

3. With respect to the sidebands themselves, more than one set of sidebands may appear if the time waveform becomes less and less sinusoidal which may indicate a more serious gear wear problem.

4. If more than one gear in a mesh has worn teeth, sidebands will be established at each of the speeds of the gears having worn teeth.

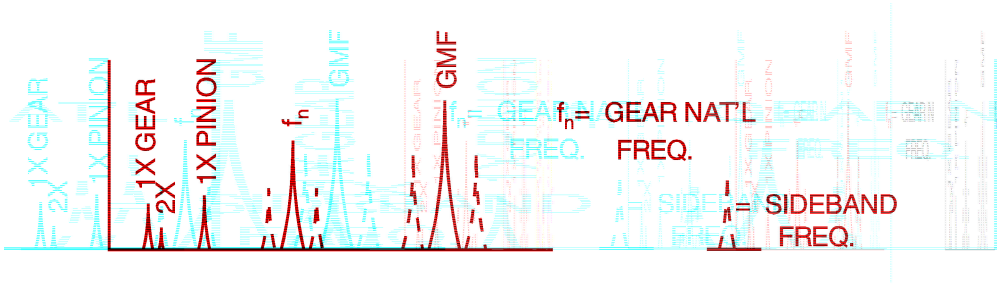

FIGURE 6.11B
SPECTRUM INDICATING GEAR TOOTH WEAR

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6.112 Significant Load Imposed on Gear Teeth:

Figure 6.11C provides a spectrum indicative of gear teeth subjected to significant load.

Gear mesh frequencies themselves are often very sensitive to the load imposed upon them. High GMF amplitudes do not necessarily indicate a problem with gear health. However, if the load is excessive and continues for a period of time, eventual fatigue of gear tooth surfaces will begin. Therefore, a gear mesh frequency which has substantially increased in amplitude between one survey and the next may not yet signify a problem (particularly if sideband amplitudes remain low and if no gear natural frequencies are excited).

Because gear mesh frequencies are sometimes so load sensitive, each vibration survey should be performed with the system under maximum operating load if this is possible.

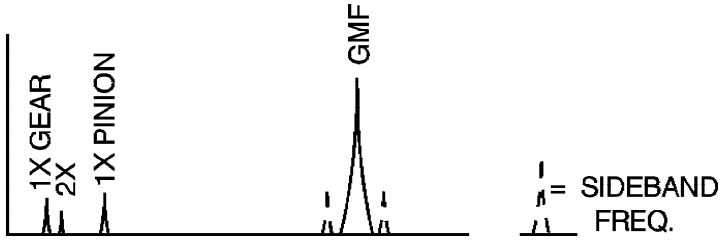

FIGURE 6.11C
SIGNIFICANT LOADING INDICATED ON GEARING

6.113 Gear Eccentricity and/or Backlash:

Figure 6.11D shows an example spectrum indicating significant gear eccentricity and/or backlash. These problems display the following characteristics:

1. Both eccentricity and backlash excite the gear natural frequencies as well as gear mesh frequency. They also may generate a number of sidebands about both the natural and gear mesh frequencies.

2. If a gear is eccentric, it will modulate the natural frequency and gear mesh frequencies, both of which will be sidebanded at 1X RPM of the eccentric gear. An eccentric gear can generate significant forces, stresses and vibration if it is forced to bottom out with the meshing gears.

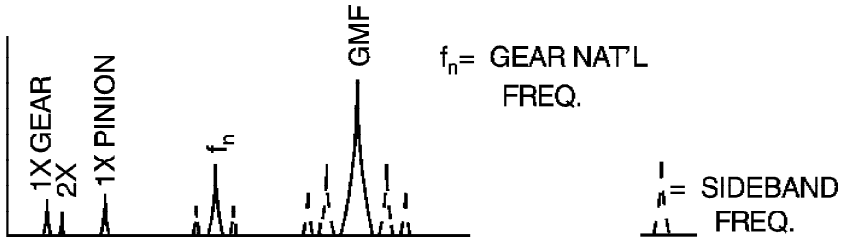

FIGURE 6.11D
GEAR ECCENTRICITY AND/OR BACKLASH INDICATED

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6.114 Gear Misalignment:

Figure 6.11E is indicative of misaligned gears which almost always excites higher order gear mesh frequency harmonics. Often, only a small amplitude will be at the fundamental gear mesh frequency (GMF), but much higher levels will be at 2X and/or 3X GMF. Often, the sideband spacing about GMF frequencies might be 2X RPM, or even 3X RPM when gear misalignment problems are involved.

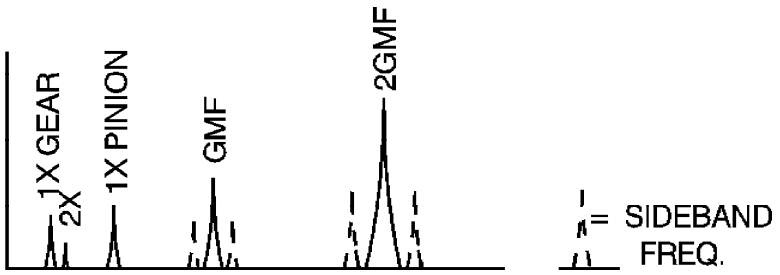

FIGURE 6.11E
SPECTRUM INDICATING MISALIGNMENT OF GEARS

6.115 Cracked, Chipped or Broken Gear Teeth:

A gear with a cracked, chipped or broken tooth will generate high vibration both at 1X RPM of this gear as well as the gear natural frequencies with sidebands around the natural frequency at gear RPM. This same behavior is exhibited by a gear tooth having a large, pronounced spall. Of course, an unbalanced gear would also cause high vibration at 1X RPM. Therefore, a time waveform like that shown in Figure 6.11F is of great assistance in determining whether the dominate problem is unbalance or gear tooth problems.

In many cases, a cracked or broken gear tooth will cause little vibration at 1X RPM in the FFT spectrum. However, it can have amplitudes 10 to 20 times higher in the time waveform with pronounced spikes spaced at the operating speed of the gear with the cracked or broken tooth.

Referring to Figure 6.11F, note that a good conditioned tooth with display a smooth, sinusoidal waveform (assuming there are no defective rolling element bearings supporting the shaft). However, cracked, chipped or broken gear teeth will generate a pronounced spike every time they go into and out of mesh. Looking at a time waveform, one can determine if the problem is with the gear teeth or from another impact event like a ball bearing problem. In the case of rolling element bearings, there would be a tremendous number of impacts within a short period of time. That is, high frequencies have correspondingly low periods. On the other hand, in the case of the gear tooth problem, if the distance in time between impact events corresponds to the RPM of the gear, this presents strong evidence of tooth problems. For example, referring to Figure 6.11F, if a 600 RPM gear showed a spike every .10 sec (.10 sec/cycle = 10 cycles/sec = 600 cycles/minute), there would be strong evidence of cracked, chipped or broken gear teeth.

If a spectrum is taken on a shaft which has more than one gear mounted on it and shows the pronounced time waveform plus natural frequencies in the spectrum sidebanded at RPM of the offending gear, it will not necessarily be clear which gear on the shaft has the problem. In this case, impulse natural frequency tests should be performed on each gear on the shaft as well as the gears to which they mesh to identify which of the gears has the problem.

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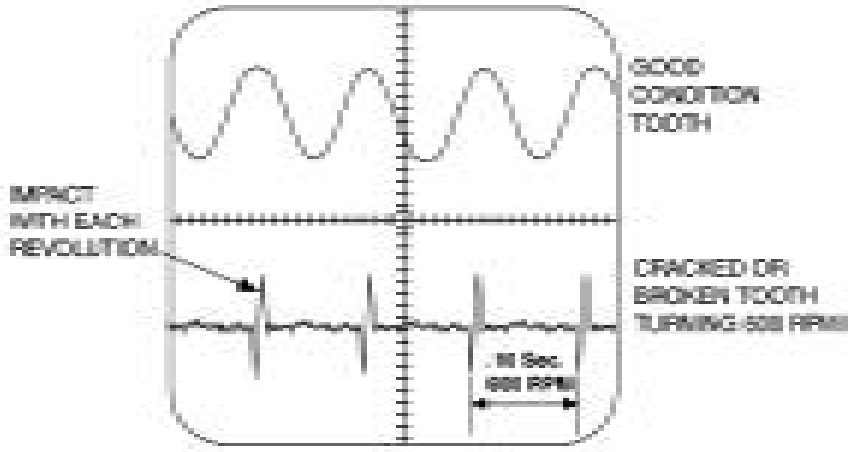

FIGURE 6.11F
COMPARISON OF TIME WAVEFORM FOR A GOOD CONDITIONED VERSUS A CRACKED OR BROKEN GEAR TOOTH

6.116 Hunting Tooth Problem:

Figure 6.11G shows a spectrum which might indicate hunting tooth problems. Hunting tooth frequencies (f_{HT}) appear when problems might have occurred during the gear manufacturing process or due to mishandling, as well as when problems occur in the field. For example, if faults appeared on both the gear and pinion, each time the defective teeth on each gear came in contact with one another, they would generate a pulse. Since most gears are not a 1:1 ratio, these two particular teeth would only come into contact periodically. For example, consider the case of a pump gerotor having a 6 tooth male and a 7 tooth female gear like that shown in Figure 6.11H. If each gear were numbered, tooth #1 on the gear would only be opposite tooth #1 on the pinion once every 7 revolutions. The actual formula for this hunting tooth frequency is given in Figure 6.11G, but is repeated here for clarity:

$$f_{HT} = \frac{(GMF)(N_a)}{(T_{GEAR})(T_{PINION})}$$

where:

f_{HT} = Hunting Tooth Frequency (Hz or CPM)

GMF = Gear Mesh Frequency
= #teeth X RPM (Hz or CPM)

N_a = Number of Unique Assembly Phases in a given Tooth Combination
(Product of Prime Factors Common to #teeth on each gear)

T_{GEAR} = # Teeth on Gear

T_{PINION} = # Teeth on Pinion

A gear set with this gear tooth repeat problem normally will generate a “growling” sound from the drive. It can cause quite high vibration, but since it occurs at low frequencies predominately below 600 CPM, it is often missed. It often requires the use of a seismic accelerometer to even detect its presence. However, left uncorrected, it can be quite destructive to gear teeth.

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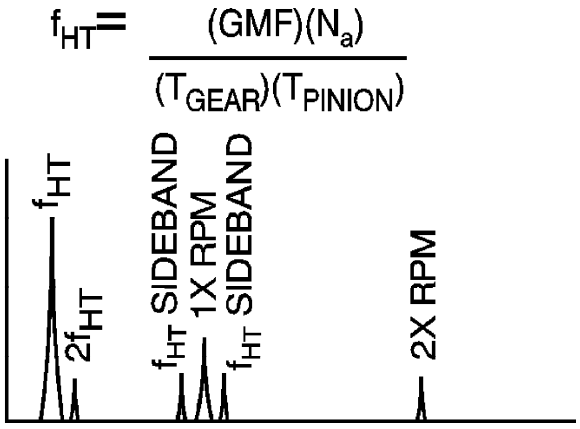
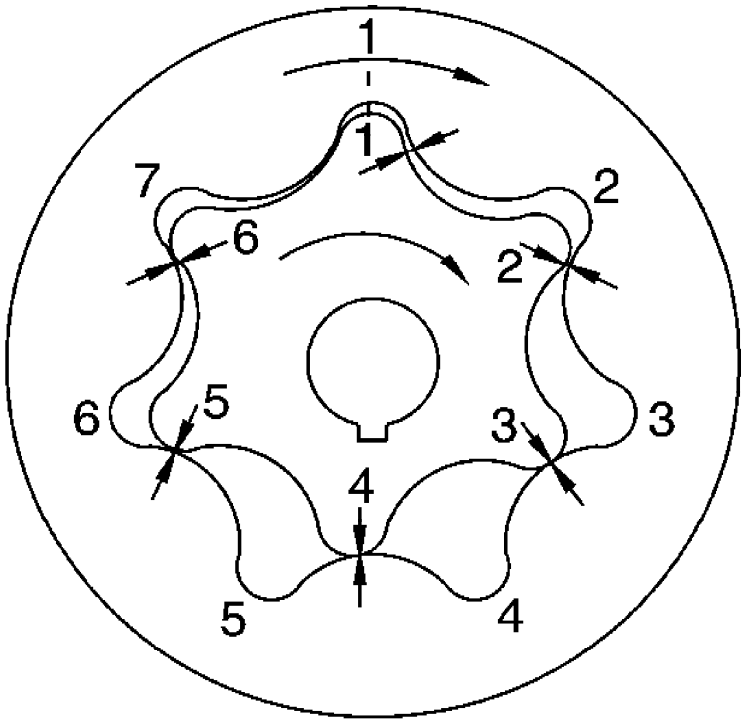


FIGURE 6.11G



Courtesy Double A Products Co.

FIGURE 6.11H
SCHEMATIC DIAGRAM OF THE GEROTOR MECHANISM

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6.12 ELECTRICAL PROBLEMS

Many find it surprising that you can detect not only mechanical problems, but also electrical problems using vibration signature analysis. However, remember that the magnetic fields within a motor create flux which induces electromagnetic forces, and that these, along with forces from mechanically induced problems must all be supported by the bearings. These forces are then measured directly by force transducers placed on the bearing housings, or indirectly by vibration transducers such as accelerometers, velocity pickups or proximity probes. Accelerometers directly measure the quotient of force divided by mass (f/m) which is acceleration (according to Newton's Law). Again, it does not matter whether these forces themselves are mechanically or electrically induced.

Figures 6.12A and 6.12B picture the stator and rotor of an induction motor, along with a close-up view of a rotor, rotor bars and shorting rings. Some of the problems which can be detected using vibration analysis include each of the following:

- Stator Eccentricity, Shorted Laminations and Loose Iron
- Eccentric Rotor (Variable Air Gap)
- Rotor Problems (Broken or Cracked Rotor Bars or Shorting Rings, Shorted Rotor Laminations, Loose Rotor Bars, etc.)
- Thermal Bow Induced By Uneven Localized Heating of a Rotor
- Electrical Phasing Problems Due to Loose or Broken Connectors
- Problems with Synchronous Motors
- Problems with DC Motors
- Torque Pulse Problems

Each of the above problems will be discussed in Sections 6.121 through 6.128 which will follow. However, a few things should be said about detection of electrical problems in general before getting into specifics.

First, it is important that if electrical problems are suspected, diagnostic tests to evaluate electrical condition should be run with the motor fully loaded. This is particularly due to the fact that the electromagnetic forces themselves vary with the square of stator current as per Reference 7. Very often, motors with proven electrical problems will not generate vibration signatures showing problem symptoms when operated "solo" (uncoupled from the driven machine), or even when operating a machine in an unloaded condition such as an air compressor. In these cases, definite symptoms appear when such machines are loaded, particularly at or near 100% load. Of course, this relates directly back to the fact that electromagnetic induced forces are so dependent on the current draw.

Probably a majority of all electrical problems detected involve "higher than normal" amplitudes at 2X electrical line frequency (also known as synchronous frequency). In the United States, the line frequency is set at 60 Hz whereas in Europe, it is at 50 Hz. Therefore, with respect to the United States, a frequency of great importance when detecting electrical problems will be that at 120 Hz (or 7200 CPM).

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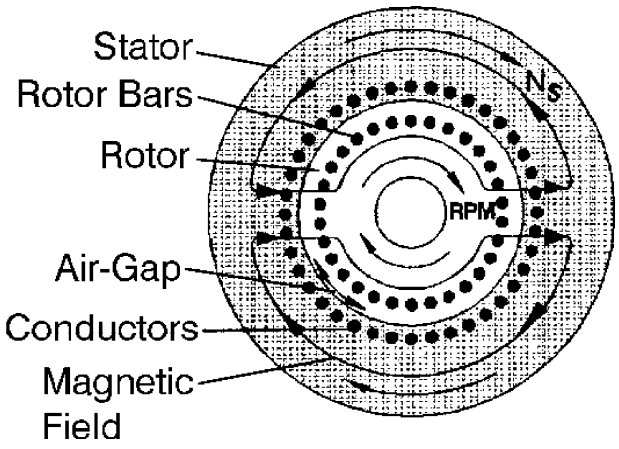

$N_S = \frac{120\ F_L}{P}$ = SYNCHRON. SPEED

$F_S = N_S$ - RPM = SLIP FREQ.

$F_P = (F_S)(P)$ = POLE PASS FREQ.

RBPF = # ROTOR BARS X RPM

WHERE:
F_L = ELECTRICAL LINE FREQUENCY (often 60 Hz)
RPM = ROTOR SPEED
N_S = SYNCHRONOUS SPEED
F_S = SLIP FREQUENCY (N_S - RPM)
F_P = POLE PASS FREQUENCY
P = # POLES
RBPF = ROTOR BAR PASS FREQUENCY

FIGURE 6.12A
DIAGRAM OF AN INDUCTION MOTOR ALONG WITH ITS STATOR, ROTOR, ROTOR BARS, AIR GAP AND MAGNETIC FIELD FOR 2-POLE MOTOR

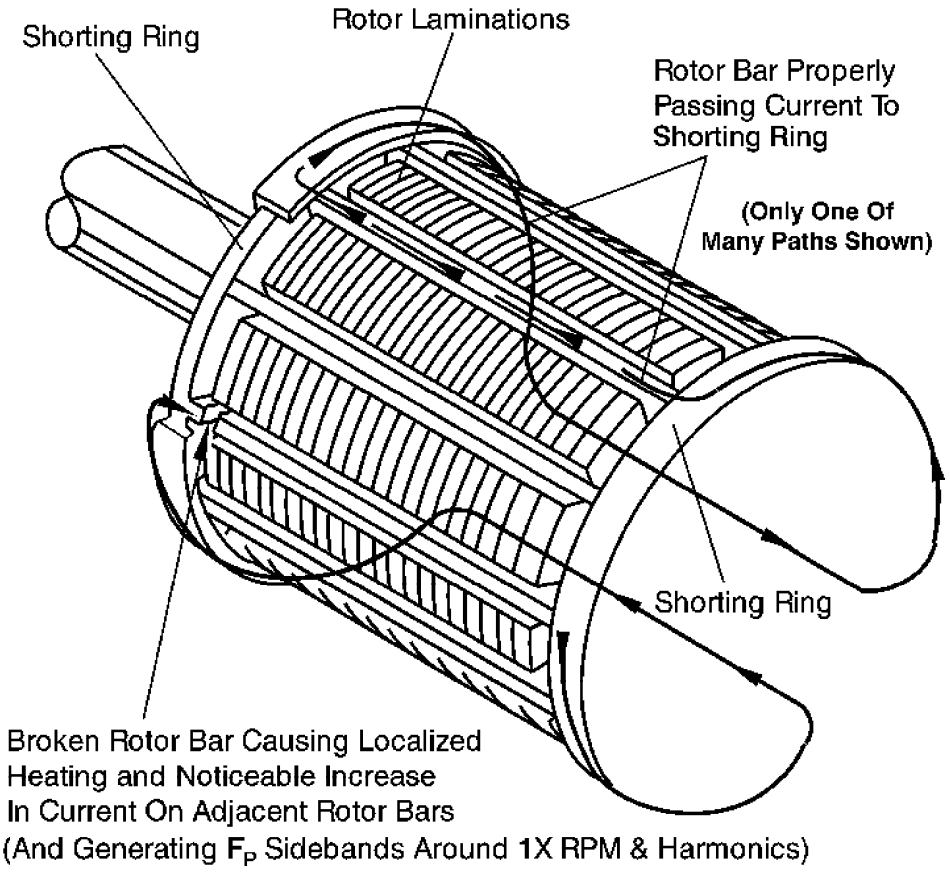

FIGURE 6.12B
ISOMETRIC VIEW OF A ROTOR INCLUDING ITS ROTOR BARS, SHORTING RINGS AND ROTOR LAMINATIONS FOR 2-POLE MOTOR

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Figure 6.12C helps explain why so many electrical problems involve twice line frequency rather than its fundamental. Referring to Figure 6.12C, note that during one rotation of the 3600 CPM stator field, the magnetic pull towards the closest pole rises from 0 to maximum twice in an eccentric rotor. Since the field itself revolves at 3600 revolutions per minute, the magnetic pull reaches a maximum 7200 times per minute (or 7200 CPM). Another way of looking at it is that because the close side of the rotor will first be attracted to the north, and then to the south pole, the force itself will vary at 2X the frequency of the magnetic field relative to the eccentricity. Therefore, when the rotor is not centered within the stator (either due to an eccentric rotor or stator), it will always affect 7200 CPM vibration. Figure 6.12D shows the difference between a stationary air gap problem where the gap itself remains at the same angular location and a rotating air gap variation which is caused by an eccentric rotor.

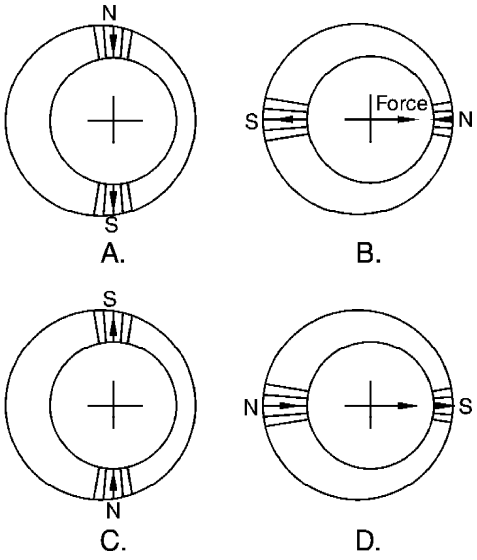

FIGURE 6.12C
UNEVEN AIR GAP BETWEEN MOTOR AND STATOR

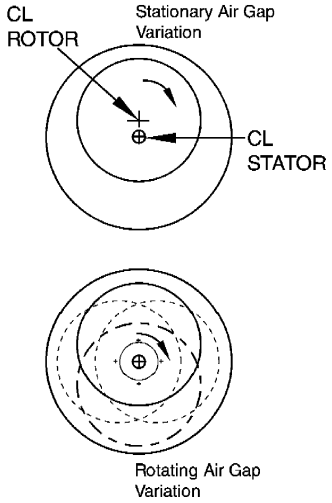

FIGURE 6.12D
STATIONARY & ROTATING AIR GAP VARIATIONS

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Before discussing diagnoses of specific electrical problems, several key items below will be defined which are common to many of the problems discussed:

(a) F_L = Electrical Line Frequency
(60 Hz = 3600 CPM in USA; 50 Hz in Europe)

(b) $N_S = \frac{120F_L}{P}$
where:

N_S = Synchronous Speed with no Slip (RPM)
P = Number of Poles

(c) $F_S = N_S$ - RPM
where:

F_S = Slip Frequency (CPM)

(d) F_P = (#Poles)(Slip Frequency)
where:

F_P = Pole Pass Frequency = a key sideband frequency which will signal the presence of many electrical problems

= $2F_S$ for 2 Pole, nominal 3600 RPM motors
= $4F_S$ for 4 Pole, nominal 1800 RPM motors
= $6F_S$ for 6 Pole, nominal 1200 RPM motors

Equation (b) is used to calculate the Synchronous Speed of a motor driven by an Inverter which can be dialed into a range of electrical line frequencies (F_L). See Table 6.12A below.

TABLE 6.12A
SYNCHRONOUS SPEEDS (RPM) FOR VARIOUS NUMBER OF POLES AT 60 Hz LINE FREQUENCY

No. of Poles	SYNCH. SPEED
2	3600
4	1800
6	1200
8	900
10	720
12	600

(e) RBPF = #Bars X RPM
where:

RBPF = Rotor Bar Pass Frequency
#Bars = The number of Rotor Bars

(f) Air Gap = Gap Between Rotor and Stator in an Electric Motor

(g) Magnetic Center = Point where Stator and Rotor Magnetic Fields are in balance and point to which rotor would be drawn.

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Figure 6.12E shows a typical spectrum which would indicate stator problems. Note the higher than normal vibration of .134 in/sec at 2X line frequency ($2F_L$). It is important to point out that when using most analyzers and data collectors, very good frequency resolution will have to be employed to separate 2X line frequency from motor running speed harmonics (2X RPM in the case of a 2 pole motor; 4X RPM in the case of a 4 pole, etc.). That is, for most frequency ranges chosen, the analyzer will not show 2 individual frequencies at 2X RPM and $2F_L$; instead, it may display only 1 frequency with a so-called value of "7200 CPM" (depending on the maximum) frequency chosen). In these cases, it will be necessary either to increase the number of FFT lines, lower F_{MAX}, or generate a "zoom spectrum" around 7200 CPM like that shown in Figure 6.12E in order to separate $2F_L$ from running speed harmonics.

For example, on Figure 6.12E which was a 2 pole motor running at 3580.5 RPM, you would not be able to separate its 2X running speed (7161 CPM) from 7200 CPM if you employed 400 FFT lines and a maximum frequency of 60,000 CPM (since you could only resolve frequencies 150 CPM apart in this case). In this case, it would be a good idea either to use a factor of 10 zoom from 12,000 CPM (giving a 3 CPM resolution) or, for example, use 3200 FFT lines (which would give a resolution of 3.75 CPM). Then, one could evaluate whether the problem is of a mechanical nature (running speed harmonics) or of electrical (2X line frequency).

Importantly, it is a good idea to use log amplitude rather than linear amplitude when performing zoom spectra for any electrical problem when it will be necessary to zoom in around line frequency, 2X line frequency, rotor bar pass frequency, running speed harmonics or fundamental pole pass frequencies themselves (F_L). If log amplitude is not employed, pole pass frequency sidebands around peaks such as 1X RPM, line frequency or 2X line frequency may easily be missed since such pole pass sidebands are normally so much lower in amplitude, and yet can still signal potential problems.

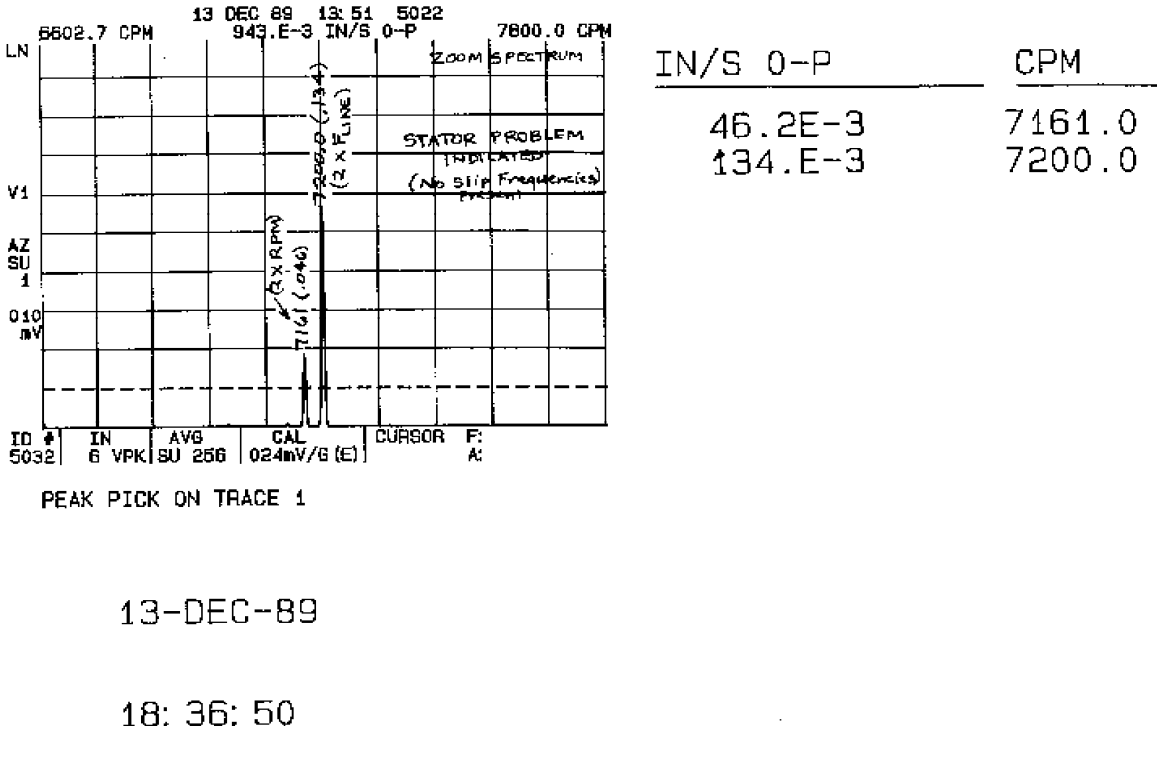

FIGURE 6.12E
ZOOM SPECTRUM INDICATING STATOR PROBLEM

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Following below will be a discussion of each of the electrical problems listed above in the introduction to the electrical problem section.

6.121 Stator Problems (see Table 6.0, Section A):

Stator problems detectable by vibration analysis include each of the following:

- Stator Eccentricity (an eccentric stator producing a stationary differential air gap between the rotor and stator)

- Shorted Laminations (insulation problems with lamination layers which can cause localized heating

- Loose Iron (any looseness or weakness in the stator)

These problems exhibit the following characteristics:

1. All such stator problems generate high vibration at 2X line frequency. However, they do not necessarily generate pole pass frequency sidebands since they originate within the stator, and are not therefore modulated by either running speed or slip frequency. Figure 6.12G indicates a serious electrical problem. Figure 6.12F is a spectrum that was captured by a data collector during a regularly scheduled PMP route. Note the high amplitude of .230 in/sec at 7200 CPM which exceeded Band 3. A real-time analyzer was then employed on the same day to better define this problem which is shown by the zoom spectrum of Figure 6.12G. This spectrum showed a high level of .228 in/sec at 7200 CPM ($2F_L$) and only .0044 in/sec at 2X RPM. Please also note the absence of any pole pass sidebands about 2X line frequency which indicated a stator problem. Upon inspection, a stator eccentricity problem was found.

2. Concern should be given motors whose vibration exceeds .050 in/sec peak at 2X line frequency on new or rebuilt motors, or .100 in/sec peak on in-service motors having a stator problem (applies to motors in general ranging from 50 HP to 1000 HP). This amplitude applies specifically to the peak at 7200.0 CPM itself ($2F_L$). However, if this motor is directly driving a precision machine tool spindle, $2F_L$ levels will have to be much lower, on the order of .025 in/sec or less.

3. Stator eccentricity produces an uneven stationary air gap between the rotor and stator which results in highly directional vibration, depending on the largest gap differential. The largest magnetic forces occur at a minimum rotor/stator gap. Therefore, the electromagnetic forces themselves go from a minimum to a maximum each revolution producing vibration at twice line frequency (7200 CPM).

4. Differential air gaps should not exceed 5% for induction motors and 10% for synchronous motors. If the vibration amplitude at $2F_L$ grows over time, the motor should be inspected (if physically possible) by marking a point both on the rotor and the stator. Then, measuring the air gap at the point where the marks align, rotate the rotor in 45° increments and measure the air gap at the point where the stator is marked. If the variation exceeds approximately 5%, the air gap difference is due to an eccentric rotor. The next step is to rotate the rotor again in 45° increments, this time measuring the gap at the point where the rotor is marked. If this gap varies more than 5% for these measurements, an eccentric stator is indicated.

5. Loose iron is due to localized stator support weakness or looseness.

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6. Shorted stator laminations can cause uneven, localized heating which can actually distort the stator causing a stator eccentricity problem. This produces thermally-induced bow which can significantly grow with operating time. The thermally induced bow can sometimes actually cause the stator to bow and contact the rotor, which can be catastrophic.

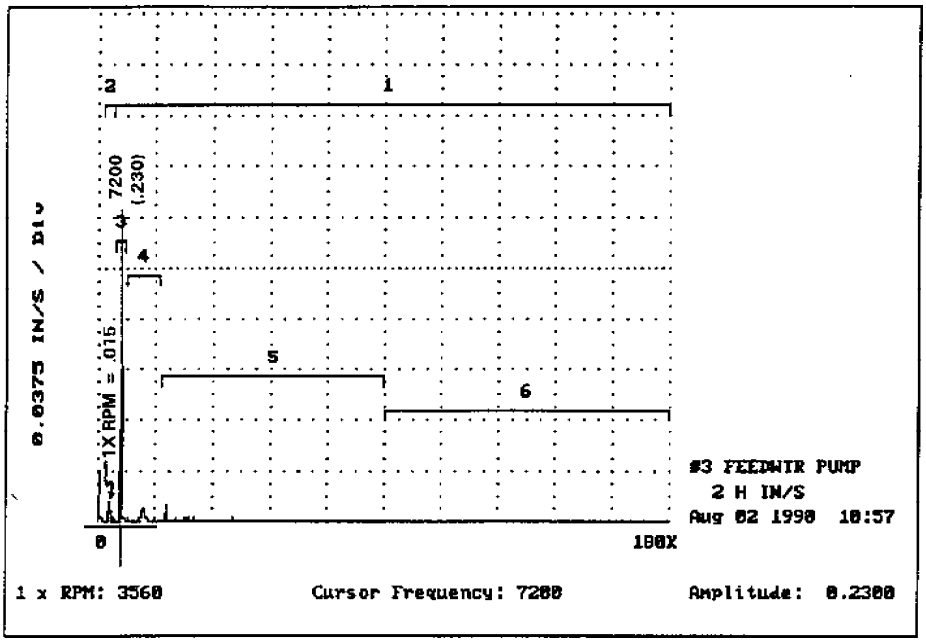

FIGURE 6.12F
NORMAL PMP ROUTE SPECTRUM WITH FMAX = 50X RPM

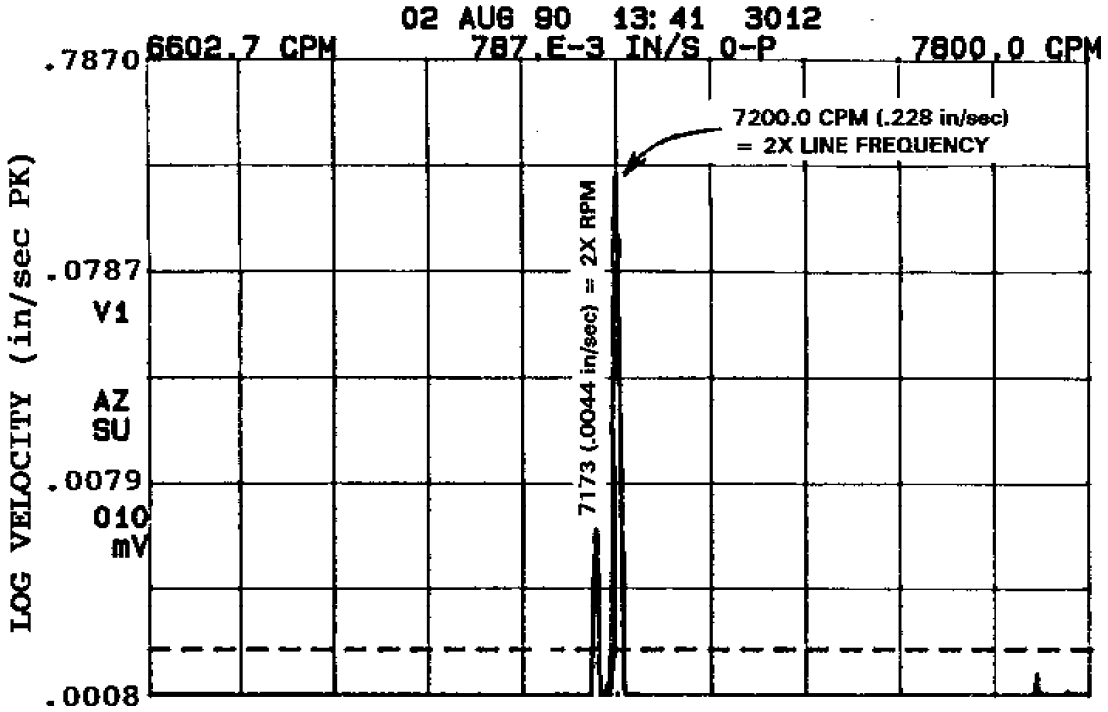

FIGURE 6.12G
ZOOM SPECTRUM INDICATING PROBLEM WITHIN THE STATOR
(NOTICE THE ABSENCE OF POLE PASS SIDEBANDS)

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6.122 Eccentric Rotor (see Table 6.0, Section B):

In the case of an eccentric rotor, the rotor itself is not concentric with its centerline (see Figure 6.12D). Therefore, an eccentric rotor produces an uneven air gap between the rotor and stator which will rotate with the rotor (as opposed to a stationary air gap in the case of an eccentric stator). An eccentric rotor can be caused by shorted rotor laminations resulting in localized heating and inducing a bowed rotor, or a simple out-of-round rotor.

1. An eccentric rotor most often will produce a high vibration at twice line frequency ($2F_L$) accompanied by sidebands spaced at pole pass frequencies (F_P = #Poles X slip frequency). That is, for a two-pole, 3600 RPM motor, the sidebands will be at twice slip frequency; while for a four-pole motor, sidebands would be at 4X slip frequency. Figure 6.12I shows a spectrum indicative of an eccentric rotor problem producing a variable air gap. First, the wideband spectrum of Figure 6.12H showed a high overall of .295 in/sec with .162 in/sec at a "so-called" 7200 CPM frequency. The zoom spectrum of Figure 6.12I revealed a high level of .166 in/sec at 7200 CPM, along with a well-formed pole pass sideband indicative of an eccentric rotor.

2. Concern should be given motors whose amplitude at twice line frequency (7200 CPM) exceeds approximately .050 in/sec for new or rebuilt motors; or .100 in/sec peak for in-service motors assuming these are general purpose motors. If serving a precision machine tool spindle, the allowable at $2F_L$ would be only approximately .025 in/sec. This would apply to most induction motors ranging in size from approximately 50 HP to 1000 HP. It is important to clarify that this amplitude applies specifically to 2X line frequency itself.

3. An eccentric rotor may require adjustment of bearing housings themselves and/or machining of the rotor journals in order to provide an air gap within tolerance all around the periphery.

4. In a predictive maintenance program, when an eccentric rotor is indicated by pole pass frequency sidebands around $2F_L$, they should be closely trended in future surveys. For example, when a peak at 2FL exceeds roughly .100 in/sec, it should be closely trended in future surveys. If its amplitude noticeably increases, and if sideband amplitudes do likewise, much greater concern should be given, particularly if even more sidebands appear above and below $2F_L$ (normally 7200 CPM). On the other hand, if the amplitudes at 7200 CPM and sidebands remain stable over several surveys, no further damage is likely being done to the motor - even if these amplitudes are as much as .175 in/sec at 7200 CPM. In these cases, it will likely be satisfactory just to continue trending. However, the motor itself may have a lowered life expectancy.

5. Note that a motor having an eccentric rotor will often experience higher and higher vibration as it comes up to temperature. For example, when such a motor is first started, it may have a level of only .10 in/sec. Then, after about 10 minutes operating time, the amplitude might increase to, for example, .14 in/sec. Next, after about 20 minutes, it may continue to grow up to about .18 in/sec. Finally, after 30 minutes, it may experience possibly .25 in/sec or more. This can be caused by nonuniform heating of the rotor on one side relative to the other. Left uncorrected, it can result in catastrophic failure if the rotor bows sufficiently to throw the rotor into the stator.

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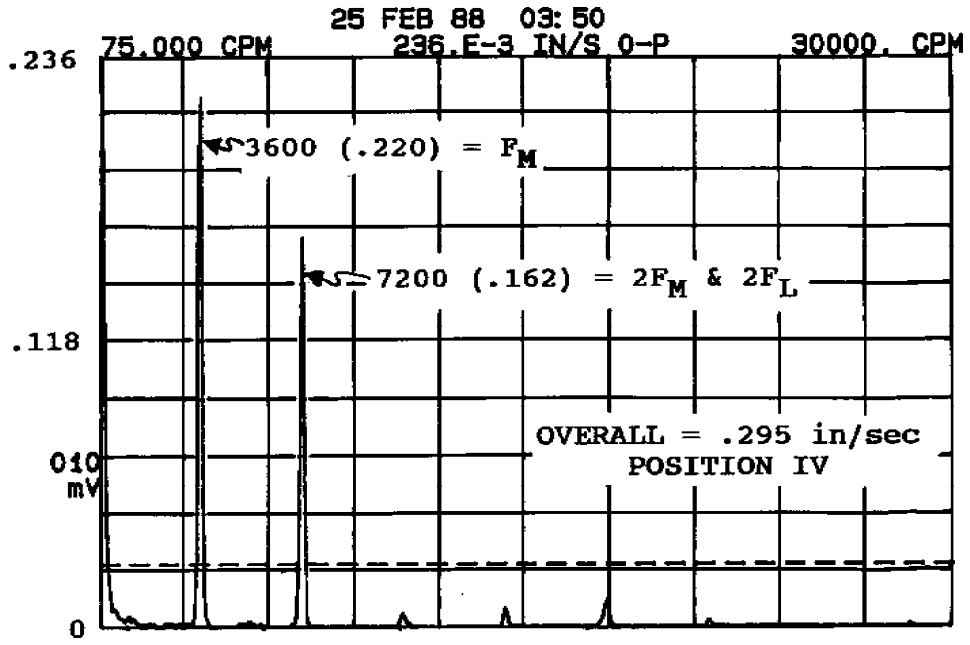

FIGURE 6.12H
30,000 CPM SPECTRUM FOR A CIRCULATING WATER PUMP MOTOR

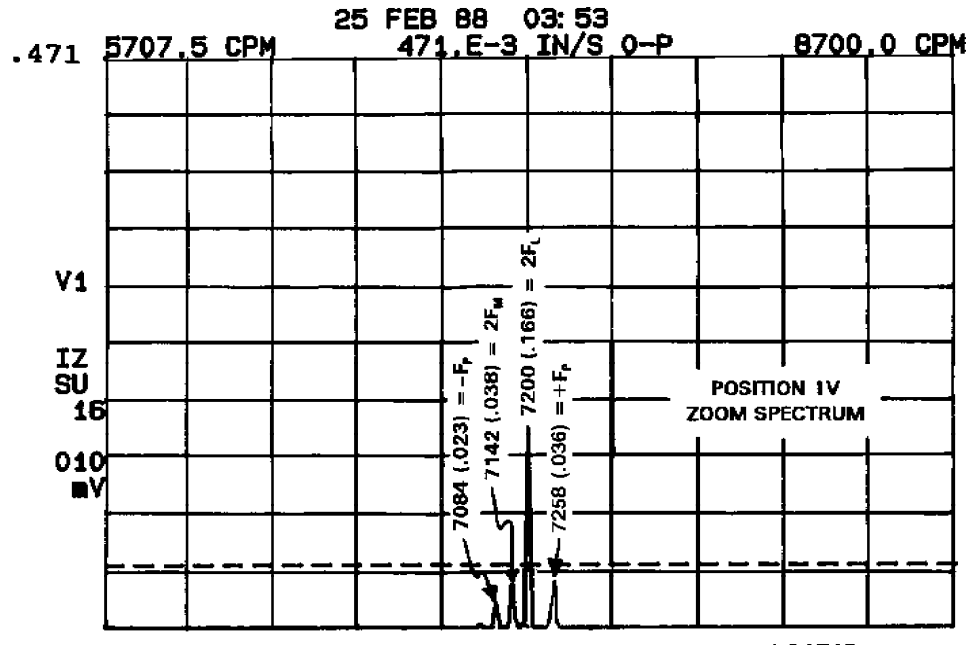

FIGURE 6.12I
ZOOM SPECTRUM INDICATING AN ECCENTRIC ROTOR
(NOT MECHANICAL LOOSENESS OR MISALIGNMENT)

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6.123 Rotor Problems (see Table 6.0, Section C):

Rotor problems detectable by vibration analysis include each of the following:

- Broken/Cracked Rotor Bars or Shorting Rings (see Figure 6.12B)
- Bad High Resistance Joints between Rotor Bars & Shorting Rings
- Shorted Rotor Laminations
- Loose/Open Rotor Bars not making good contact with End Rings

These problems exhibit the following characteristics:

1. Probably the key area of concern for broken or cracked rotor bars, is the presence of pole pass frequency sidebands around 1X RPM (in the case of 2-pole motors, these sidebands will be at 2X slip frequency while at 4X slip frequency for 4-pole motors). Figures 6.12J thru 6.12M show spectra typical for a 2-pole motor having serious rotor problems. Initially, the 30,000 CPM wideband spectrum gave little hint of a serious problem, seemingly showing mechanical looseness as evidenced by several running speed harmonics. However, upon closer examination, the zoom spectra of Figures 6.12K thru 6.12M revealed a series of well-formed pole pass sidebands around 1X, 2X and 3X RPM, respectively. These spectra all indicated cracked or broken rotor bars, shorting ring problems or shorted rotor laminations.

2. As suggested by Figures 6.12K thru 6.12M, in addition to the pole pass frequency sidebands around 1X RPM, broken or cracked rotor bars and/or high resistance joints can produce pole pass sidebands around higher running speed harmonics up to and including the 2nd, 3rd, 4th and 5th running speed harmonics. In this case, more than 1 rotor bar is often found cracked or broken since there is more than 1 pulse event per revolution. Figure 6.12N shows such a spectrum with multiple pole pass sidebands around 1X, 2X, 3X, and 4X RPM. In this case, the operating speed was about 1176 RPM (meaning that slip frequency = F_S = 24 CPM). Since this was a 6-pole motor, the pole pass frequency (F_P) equalled 6X 24 CPM = 144 CPM. This was the approximate spacing of each of the multiple F_P sidebands about 1X RPM through 5X RPM as shown by the zoom spectrum of figures 6.12P. Later, this motor was found to have 4 cracked rotor bars.

3. The key area of concern for loose/open rotor bars is vibration at much higher frequencies at rotor bar pass frequency (RBPF) and also harmonics of this frequency (RBPF = # Rotor Bars X RPM).

4. Here, the concern is amplitudes exceeding approximately .06 in/sec at either rotor bar pass frequency (RBPF) or higher harmonics (2 RBPF or 3 RBPF). In addition, the sideband spacing around RBPF and its harmonics will be exactly twice line frequency ($2F_L$). The reader is cautioned that while RBPF itself may be acceptable, if he extends the frequency range to encompass 2X RBPF, he may find amplitudes 10 or more times those at RBPF.

   For example, Figure 6.12Q shows a spectrum for a motor confirmed to have two or more open rotor bars. This motor had 57 rotor bars and operated at a speed of 1793 RPM, giving the fundamental RBPF at about 102,200 CPM. The wideband spectrum of Figure 6.12Q showed an amplitude of only .008 in/sec at RBPF. However, the story out at 2X RBPF was completely different. The zoom spectrum of Figure 6.12Q showed an excessive .340 in/sec at 204,380 CPM, or 2X RBPF (over 28 times higher amplitude than at RBPF). Importantly, if a maximum frequency high enough only to capture the fundamental RBPF was taken, this problem would have been missed entirely. Again, the key indicators were the excessively high level at 2X RBPF, which was accompanied by sidebands at exactly 7200 CPM ($2F_L$).

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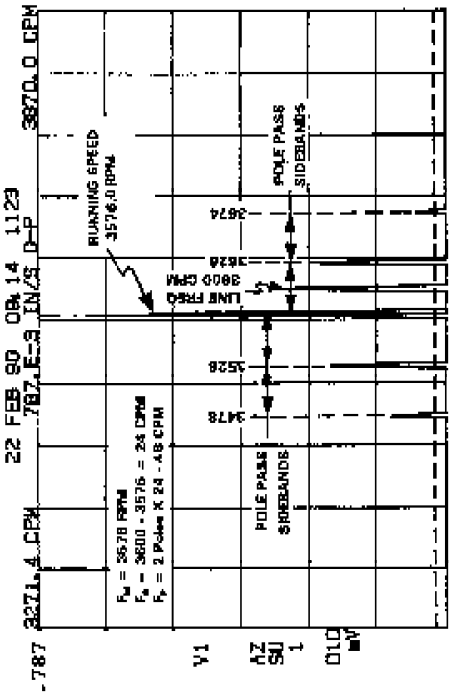

FIGURE 6.12K
ZOOM SPECTRUM AROUND 1X RPM SHOWING MULTIPLE POLE PASS SIDEBANDS (BROKEN OR CRACKED ROTOR BARS/SHORTING RING PROBLEMS INDICATED)

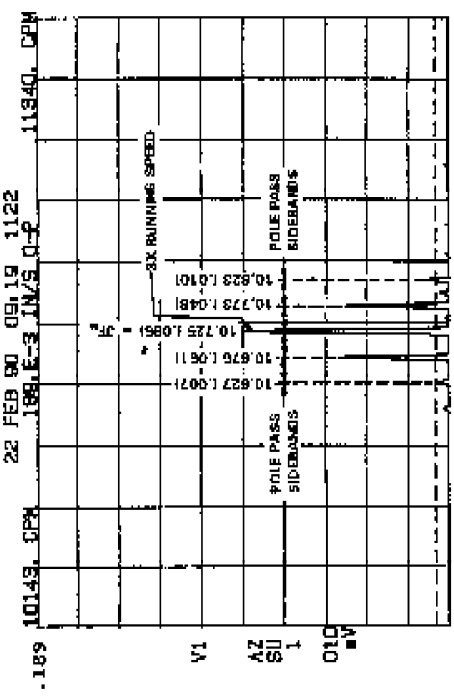

FIGURE 6.12M
ZOOM SPECTRUM AROUND 3X RPM
(NOTE POLE PASS SIDEBANDS HERE ALSO)

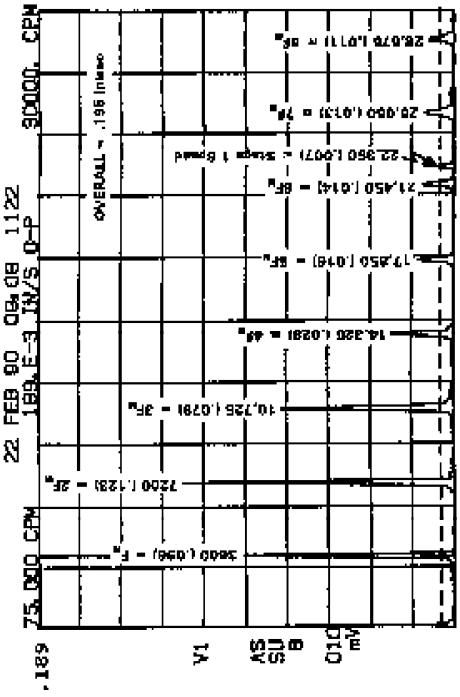

FIGURE 6.12J
30,000 CPM SPECTRUM ON A CENTAC MOTOR (POS.2H)

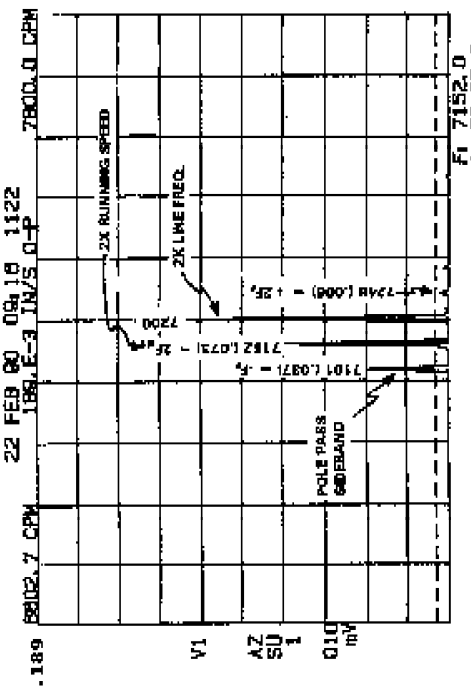

FIGURE 6.12L
ZOOM SPECTRUM AROUND 2X RPM
(POLE PASS SIDEBANDS ALSO HERE)

WIDEBAND & ZOOM SPECTRA FOR A MOTOR HAVING CRACKED OR BROKEN ROTOR BARS OR SHORTING RING PROBLEMS

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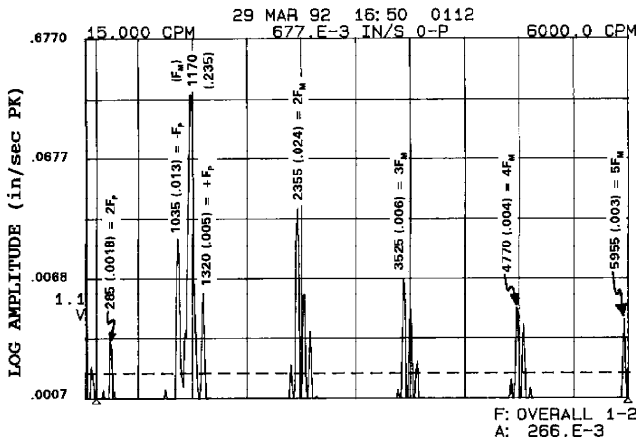

FIGURE 6.12N
LOG AMPLITUDE SPECTRUM CLEARLY SHOWING POLE PASS FREQUENCY SIDEBANDS AROUND 1X RPM THRU 4X RPM (4 CRACKED ROTOR BARS WERE FOUND IN MOTOR)

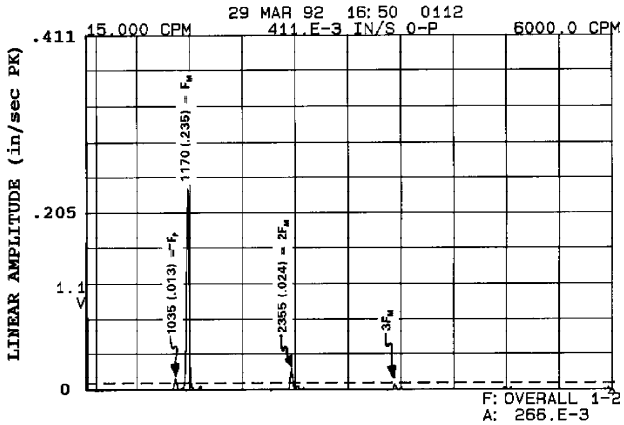

FIGURE 6.12O
LINEAR AMPLITUDE SPECTRUM DOES NOT ADEQUATELY SHOW MULTIPLE POLE PASS FREQUENCY SIDEBANDS

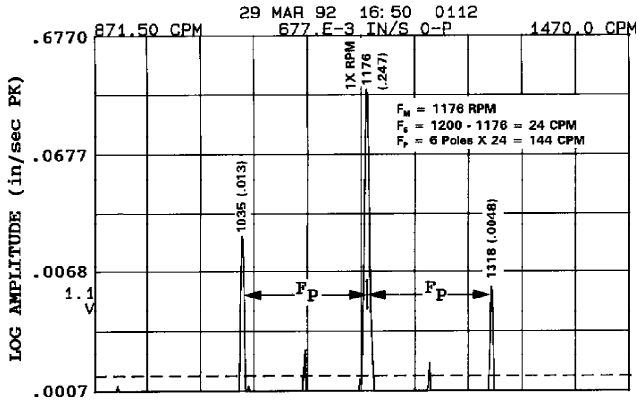

FIGURE 6.12P
LOG AMPLITUDE ZOOM SPECTRUM CLEARLY DISPLAYS 1X RPM AND POLE PASS SIDEBANDS

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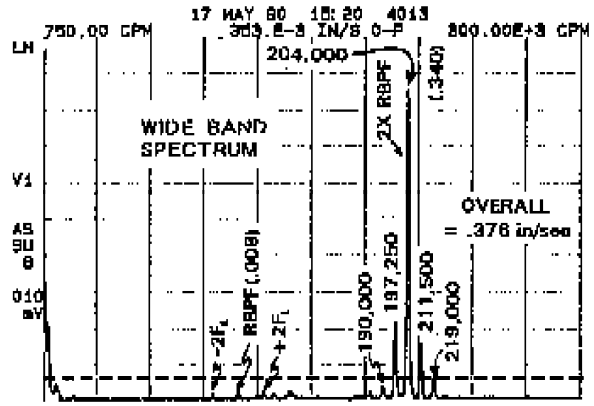
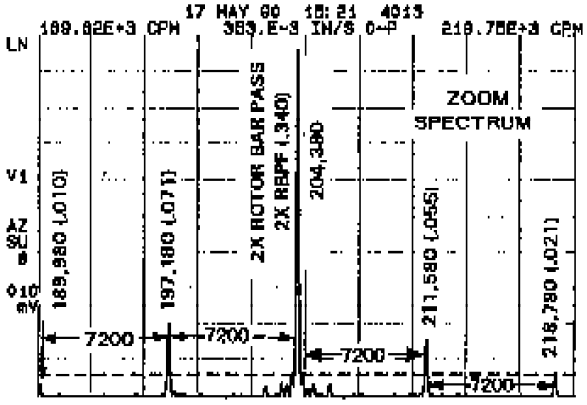


IN/S 0-P	CPM
77.4E-3	3000.0
77.4E-3	197.24E+3
305.E-3	204.00E+3
56.4E-3	211.49E+3
21.0E-3	219.00E+3

SERIOUS ROTOR BAR PROBLEM DETECTED AT 2X RBPF, BUT COMPLETELY MISSED BY 1X ROTOR BAR PASS

FINISH FAN



IN/S 0-P	CPM
70.9E-3	197.18E+3
21.0E-3	204.07E+3
340.E-3	204.38E+3
21.0E-3	204.60E+3
54.8E-3	211.58E+3

SERIOUS ROTOR BAR PROBLEM DETECTED AT 2X ROTOR BAR PASS FREQUENCY

NOTE: 6-10-90 G.E. Repair Shop phoned to confirm this motor had 2 or more open rotor bars which will likely result in motor replacement.

FIGURE 6.12Q
SERIOUS ROTOR BAR PROBLEM DETECTED AT 2X ROTOR BAR PASS FREQUENCY

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5. Referring back to the lower spectrum on the Diagnostic Chart in Table 6.0 under "Rotor Problems", note that sometimes the amplitudes of RBPF or its harmonic will not be the highest amplitude. Instead, one of the sidebands spaced at difference frequencies of 2X line frequency (usually 7200 CPM with 60 Hz F_L) might be highest. This "array" of frequencies will still include RBPF and exactly $2F_L$ sidebands, and will still indicate loose or open rotor bars (assuming 60 Hz F_L) and/or variable air gap.

6. Even though the amplitude of concern in the area of rotor bar pass frequency and harmonics is about .06 in/sec for most motors, the important task will be first to detect these problems, and then to trend them before making any rash decisions about overhauling motors. If trending of several sets of spectra do not show any real increases, substantial damage is likely not continuing even with RBPF levels of approximately .10 to .15 in/sec. On the other hand, if the rate of change shows substantial increases from one survey to the next, it does indicate rapidly deteriorating condition which does warrant maintenance action.

7. In Condition Monitoring programs, it is a good idea to specify each of the following two special points on a route for each motor to detect electrical problems:

a. Low Frequency Motor Electrical Point:

   F_{MAX} = 12,000 CPM; 3200 FFT lines; 2 Ave. This will allow one to separate the true amplitude at $2F_L$ and at motor running speed harmonics.

b. High Frequency Rotor Bar Pass Evaluation Point:

   F_{MAX} = 360,000 CPM; 1600 FFT lines; 8 Ave. (For motors with more than two poles, an F_{MAX} of 240,000 CPM will likely be sufficient to pick up both 1X and 2X RBPF.) This will allow the detection of potential problems at RBPF and its multiples. Look for difference frequencies spaced at exactly $2F_L$ (usually 7200 CPM), even if the number of rotor bars is unknown. Figure 6.12R is a good example. Note the high amplitude of .136 in/sec at RBPF which penetrated both "Alarm 1" (.060 in/sec) and "Alarm 2" (.100 in/sec) narrowband envelopes. Also, notice the 7200 CPM ($2F_L$) sidebands surrounding RBPF. The reader is cautioned to carefully mount his transducer and to employ one with sufficient frequency response out to 360,000 CPM (6000 Hz) if he must make these higher frequency measurements on 2-pole motors.

Experience has shown that these measurements are normally best taken in the horizontal direction. Remember that these two spectral measurements are in addition to the standard condition monitoring route points on each motor. However, they will likely prove invaluable in detecting potentially significant electrical problems before they result in catastrophic motor failure.

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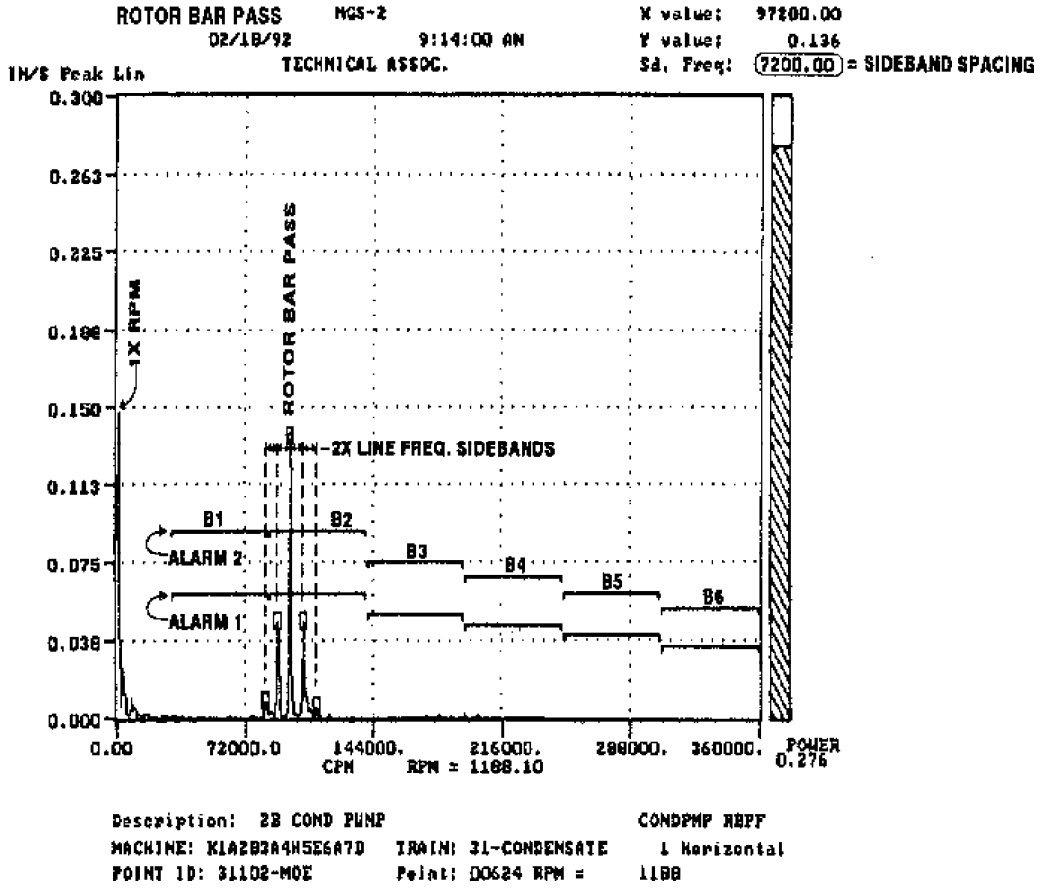

Threshold Report

PLANT: NGS-2
Description: 2B COND PUMP
MACHINE: K1A2B3A4H5E6A7B
TRAIN: 31-CONDENSATE
POINT ID: 31102-MOE
Point: 00624
Units: IN/S Peak Lin
RPM: 1188.10
Threshold: 0.005

Frequency Hz	CPM	Order	Spect Amp		% Exceeded
18.95	1136.9	0.96	0.148		2861.08
61.43	3685.6	3.10	0.024		374.55
138.02	8281.4	6.97	0.010		95.67
1383.46	83007.6	69.87	0.008		68.81
1503.55	90213.2	75.93	0.047		835.97
1623.67	97420.2	82.00	0.136		2610.03
1743.41	104604.8	88.04	0.047	ROTOR	835.97
1863.95	111836.9	94.13	0.005	BAR PASS	5.24

Power	Sync	Subsync	Non-Sync
0.276	0.262	0.000	0.087

FIGURE 6.12R
TYPICAL SETUP OF SPECTRAL ALARM BANDS FOR A 6-POLE CONDENSATE PUMP MOTOR (1180 RPM NAMEPLATE SPEED)

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6.124 Thermal Bow Induced By Uneven Localized Heating of a Rotor:

Figure 6.12S illustrates a rotor subjected to localized heating which can result in thermal bow. Grossly uneven localized heating of a rotor can occur if several adjacent rotor laminations are shorted together. This heat is generated by noticeably higher eddy current and hysteresis losses in the shorted laminations. Likewise, broken rotor bars can generate significant heat resulting in such localized heating of the rotor.

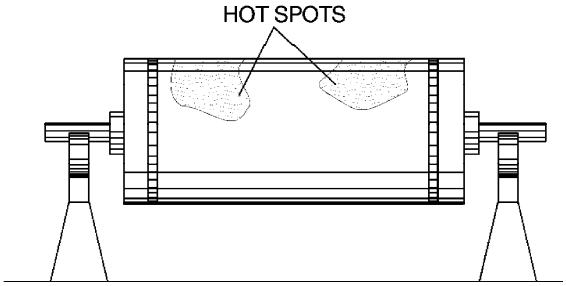

FIGURE 6.12S
DEVELOPMENT OF HOT SPOTS ON A ROTOR DUE TO BROKEN ROTOR BARS OR SHORTED ROTOR LAMINATIONS

This localized heating can actually bow the rotor which can eventually contact the stator. One of the problems with thermal bow is that it can tend to "feed on itself". That is, once it begins, the shaft will bow slightly. As the shaft bows, it will create more and more electromagnetic, unbalance forces which will create more heat. This will then cause more bow in the rotor which will again generate more heat, etc. Therefore, this problem can be catastrophic. Thermal bow is indicated by each of the following characteristics:

1. When thermal bow occurs in a rotor, one will notice that after first starting up the machine, the amplitude at 1X RPM will continue to increase with operating time as the thermal bow itself increases. It will resemble a spectrum of unbalance. However, simple balancing will have no effect.

2. As the thermal bow increases, vibration spectra will sometimes indicate eccentric rotor with high 7200 CPM vibration accompanied by slip frequency sidebands (assuming 60 Hz F_L).

3. As the thermal bow problem becomes more pronounced, vibration phase in the axial direction will approach 180° difference between that on the outboard and inboard bearings. In other words, it will simulate a bent shaft behavior.

4. An infrared camera can sometimes be used to locate the hot spots on a rotor subjected to localized heating which will be of use in determining which areas need repair.

6.125 Electrical Phasing Problems (Loose Connectors):

A spectrum indicating electrical phasing problems is included in the Table 6.0 on page 3 of the Illustrated Vibration Diagnostic Chart. Phasing problems due to loose or broken connectors can cause highly excessive vibration sometimes exceeding 1.0 in/sec at 2X line frequency ($2F_L$) which will have sidebands surrounding it spaced at one third line frequency ($1/3F_L$). This is sometimes particularly a problem if the defective connector is only sporadically making contact.

Figures 6.12T through 6.12V illustrate a severe electrical phasing problem. Figure 6.12T was captured from the outboard bearing of a motor driving a refrigeration chiller. This motor was served by an inverter which allowed it to change speed according to the incoming electrical line frequency and voltage. Figure 6.12T was captured during a normal Predictive Maintenance

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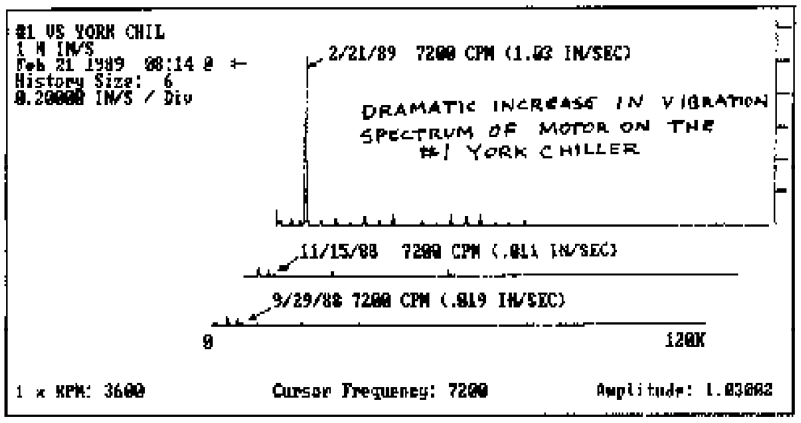
#1 VS YORK CHIL
1 N IN/S
Feb 21 1989 08:14 @ ←
History Size: 6
0.20000 IN/S / Div
2/21/89 7200 CPM (1.03 IN/SEC)
DRAMATIC INCREASE IN VIBRATION SPECTRUM OF MOTOR ON THE #1 YORK CHILLER
11/15/88 7200 CPM (.811 IN/SEC)
9/29/88 7200 CPM (.819 IN/SEC)
0
120K
1 x RPM: 3600
Cursor Frequency: 7200
Amplitude: 1.03002

3X x 2.0000 = 0.06000 1 x RPM: 3600.00

Machine	Pos	Dir	Units	Date	
#1 VS YORK CHIL	1	H	IN/S	Feb 21 1989	08:14

Frequency	1 x RPM	Amplitude	
300	0.08	0.10003	
7200	2.00	1.03002	← 2/21/89
21600	6.00	0.04800	
28500	7.92	0.06990	
42900	11.92	0.04790	
46800	13.00	0.06605	
50100	13.92	0.06990	

001X x 2.0000 = 0.02000 1 x RPM: 3600.00

Machine	Pos	Dir	Units	Date	
#1 VS YORK CHIL	1	H	IN/S	Nov 15 1988	13:09

Frequency	1 x RPM	Amplitude
3600	1.00	0.05274
6000	1.67	0.03272
21300	5.92	0.03397
49800	13.83	0.03781

000X x 2.0000 = 0.00000 1 x RPM: 3600.00

Machine	Pos	Dir	Units	Date	
#1 VS YORK CHIL	1	H	IN/S	Sep 20 1988	07:10

Frequency	1 x RPM	Amplitude	
300	0.08	0.00731	
900	0.25	0.00699	
3600	1.00	0.04988	
6000	1.67	0.03590	
7200	2.00	0.01870	← 9/29/88
10800	3.00	0.01334	
21300	5.92	0.01843	
35700	9.92	0.01176	

FIGURE 6.12T

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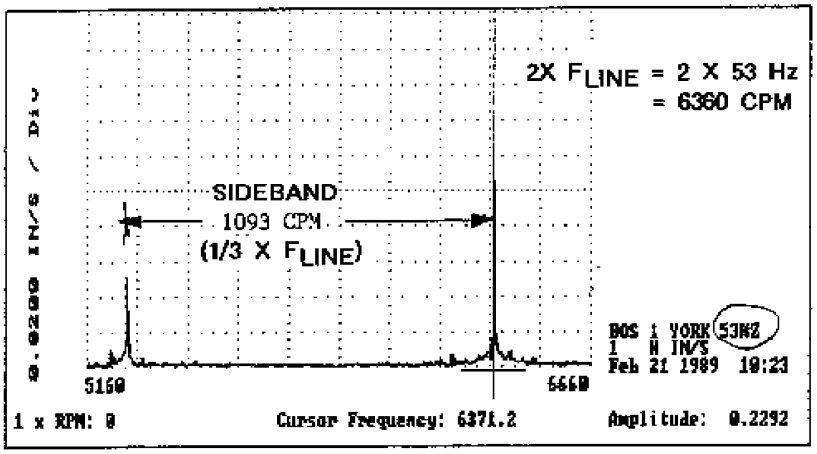
Survey by technicians from our company. Note the dramatic change that occurred at 2X line frequency on the survey of 2/21/89 increasing from .011 in/sec before up to 1.03 in/sec on 2/21/89. After consultation with engineers at our office, it was decided to lower the inverter frequency from 60 Hz to 53 Hz to see if the problem would remain, and what effect this would have on its spectral characteristics. Figure 6.12U shows a 3200 line FFT "zoom" around 2X line frequency which was now at 6371 CPM (106.2 Hz = 2X 53.1 Hz line frequency). Here, the important thing was the spacing of the sideband which was at about 1093 CPM (18.2 Hz) that placed it at 1/3 line frequency. A diagnostics of electrical phasing problems was made and recommendations were suggested to closely inspect all the supply lines and connectors, not only from the substation into the main supply, but also from the main supply into the inverter, and from the inverter into the motor itself. Subsequent checks found that one of the 3 connectors carrying one of the phases from the inverter directly into the motor itself was loose, making only sporadic contact from one instant to the next. Figure 6.12V shows the resultant effect on vibration after this loose connector was repaired. Note that the level at twice line frequency (7200 CPM) dropped dramatically from 1.03 in/sec down to .017 in/sec. It has since remained at this level during all subsequent Predictive Maintenance Surveys.



004% x 0.200 = 0.0080 1 x RPM: ??????

Machine	Pos	Dir	Units	Date
BOS 1 YORK 53HZ	1	H	IN/S	Feb 21 1989 10:23

Frequency	1 x RPM	Amplitude
5227.5	-	0.0098
5276.2	-	0.0509
6363.7	-	0.0166
6371.2	-	0.2292
6423.7	-	0.0089

FIGURE 6.12U
SIDEBANDS AT 1/3 LINE FREQUENCY DUE TO ELECTRICAL PHASING PROBLEMS

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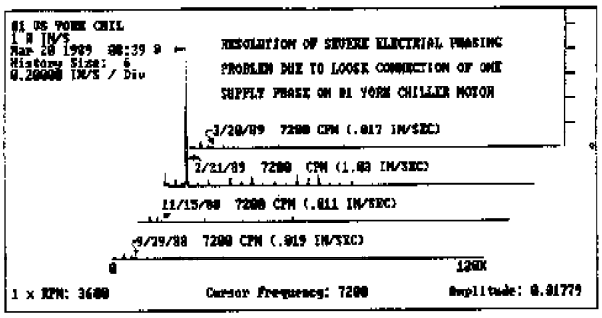
#1 VS YORK CHIL
1 X IN/S
Mar 20 1989 08:39 0
History Size: 6
0.20000 IN/S / Div
RESOLUTION OF SEVERE ELECTRICAL PHASING
PROBLEM DUE TO LOOSE CONNECTION OF ONE
SUPPLY PHASE ON #1 YORK CHILLER MOTOR
3/20/89 7200 CPM (.017 IN/SEC)
2/21/89 7200 CPM (1.83 IN/SEC)
11/15/88 7200 CPM (.811 IN/SEC)
9/29/88 7200 CPM (.919 IN/SEC)
0
126X
1 x RPM: 3600
Cursor Frequency: 7200
Amplitude: 0.01779

001% x 2.0000 = 0.02000 1 x RPM: 3600.00

Machine	Pos	Dir	Units	Date
#1 VS YORK CHIL	1	H	IN/S	Mar 20 1989 08:39

Frequency	1 x RPM	Amplitude
3600	1.00	0.06068
6000	1.67	0.02667
10800	3.00	0.02192
28500	7.92	0.02383
50100	13.92	0.02288

004% x 2.0000 = 0.08000 1 x RPM: 3600.00

Machine	Pos	Dir	Units	Date
#1 VS YORK CHIL	1	H	IN/S	Feb 21 1989 08:14

Frequency	1 x RPM	Amplitude
300	0.08	0.10008
7200	2.00	1.03002

001% x 2.0000 = 0.02000 1 x RPM: 3600.00

Machine	Pos	Dir	Units	Date
#1 VS YORK CHIL	1	H	IN/S	Nov 15 1988 13:09

Frequency	1 x RPM	Amplitude
3600	1.00	0.05274
6000	1.67	0.03272
21300	5.92	0.02097
49800	13.83	0.03781

000% x 2.0000 = 0.00000 1 x RPM: 3600.00

Machine	Pos	Dir	Units	Date
#1 VS YORK CHIL	1	H	IN/S	Sep 20 1988 09:35

Frequency	1 x RPM	Amplitude
300	0.08	0.00731
900	0.25	0.00699
3600	1.00	0.04988
6000	1.67	0.03590
7200	2.00	0.01875
10800	3.00	0.01334
21300	5.92	0.01843
35700	9.92	0.01176

FIGURE 6.12V

6.126 Synchronous Motors (Loose Stator Coils):

Figure 6.12W shows a spectrum for a synchronous motor having a problem with loose stator coils. This motor was driving a reciprocating air compressor and had experienced several bearing failures at intervals of only about 6 months. As Figure 6.12W shows, this spectrum resulted in a high vibration of .134 in/sec at about 43,200 CPM. A zoom spectrum showed that this corresponded to exactly 84X RPM. Further investigation revealed that this synchronous motor had a total of 14 poles with 6 coils per pole. Therefore, the coil pass frequency (CPF) equaled 14 poles X 6 coils/pole = 84X RPM. Note that the zoom spectrum showed not only high amplitude at coil pass frequency, but also sidebands spaced at 1X RPM of the motor.

Here, the dominant problem was diagnosed to be loose stator coils. The "After" spectrum on Figure 6.12X shows the significant improvement made by repairing the poles and filling in all accessible voids between each of the coils dropping CPF from .134 in/sec to .059 in/sec.

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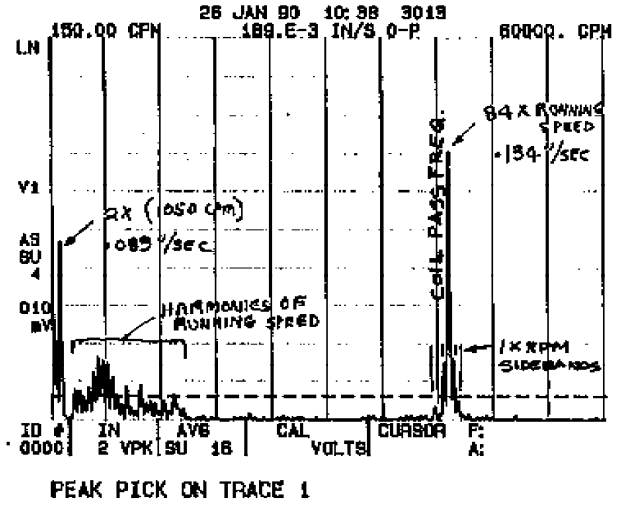


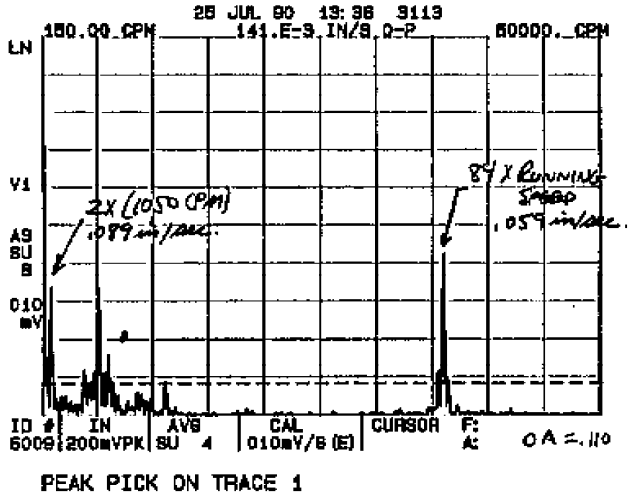
IN/S 0-P.	CPM
40.6E-3	450.00
88.9E-3	1050.0
14.7E-3	2550.0
13.8E-3	3000.0
15.5E-3	4049.9
23.3E-3	4650.0
31.1E-3	5100.0
29.3E-3	5700.0
28.5E-3	6149.7
20.7E-3	6749.7
12.1E-3	7200.0
15.5E-3	8250.0
16.4E-3	9750.0
12.1E-3	13349.
28.5E-3	42600.
134.E-3	43198.
30.2E-3	43650.

29-JAN-90

14: 11: 57

FULL LOAD - 450 AMPS
COMPRESSOR NO. 3
POS 1 VERT
JAN 26, 1990

FIGURE 6.12W
LOOSE COILS IN SYNCHRONOUS MOTOR (BEFORE SPECTRUM)
(14 POLES X 6 COILS/POLE = 84 X RPM)



IN/S 0-P.	CPM
21.9E-3	450.00
47.6E-3	1050.0
16.7E-3	4650.0
11.6E-3	5100.0
16.1E-3	5700.0
47.0E-3	6149.7
15.5E-3	6749.7
22.5E-3	7200.0
12.2E-3	7650.0
12.2E-3	13349.
16.1E-3	42748.
59.9E-3	43198.
13.5E-3	43650.

26-JUL-90

14: 27: 31

FULL LOAD - 420 AMPS
COMPRESSOR NO. 3
POS 1 VERTICAL
JAN 25, 1990

FIGURE 6.12X
RESOLUTION OF LOOSE COIL PROBLEM (AFTER REPAIR)

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6.127 DC Motor Problems:

Figure 6.12Y shows the standard internal components of a common DC motor. Many DC motor and control problems can be detected through vibration analysis. A detailed discussion on DC motor construction and this functioning of each of its controls is beyond the scope of this section but is taught in a higher level seminar (Level III). However, the purpose here is to introduce the analyst to the fact that he can use vibration analysis to detect a series of problems, not only within the DC motor itself, but also with the controls serving it. Interestingly, when a DC motor is powered by a silicon controlled rectifier (SCR) circuit, many of the vibration problems found are actually SCR firing card or other control problems. SCR circuits have come into wide use throughout industry as a method to produce DC electricity from AC electricity. SCR circuits are either half-wave rectified with 3 SCR's or full-wave rectified with 6 SCR's, with each SCR creating a pulse at a rate of 3600 CPM, or 60 Hz (see Figure 6.12Z). A 3 SCR circuit electronically processes the 3 time waveforms of 3-phase 60 Hz (in U.S.) industrial electricity into a single waveform with only the positive AC peaks remaining. The result for a 3 SCR circuit is DC electricity with a 180 Hz (10,800 CPM) pulsing rate (i.e., 3 SCR's X 3600 = 10,800 CPM). A 6 SCR circuit processes both the positive and negative peaks of the alternating 3 phase waveform into a single DC waveform with 360 Hz (21,600 CPM) pulsing rate (i.e., 6 SCR's X 3600 CPM = 21,600 CPM) These pulsing rates of 10,800 CPM (3 SCR's) and 21,600 CPM (6 SCR's) affect the rotation of the motor armature and are detectable through vibration analysis. These pulsing rates are referred to as the SCR firing frequencies. Other characteristic vibration from SCR circuit problems can result in the appearance of line frequency harmonics of F_L and harmonics of the SCR firing frequency (F_L = 60 Hz in U.S.). Amplitudes exceeding about .100 in/sec pk at the SCR firing frequency are usually considered excessive (although there are exceptions depending on size, operating speed, mounting method, etc.).

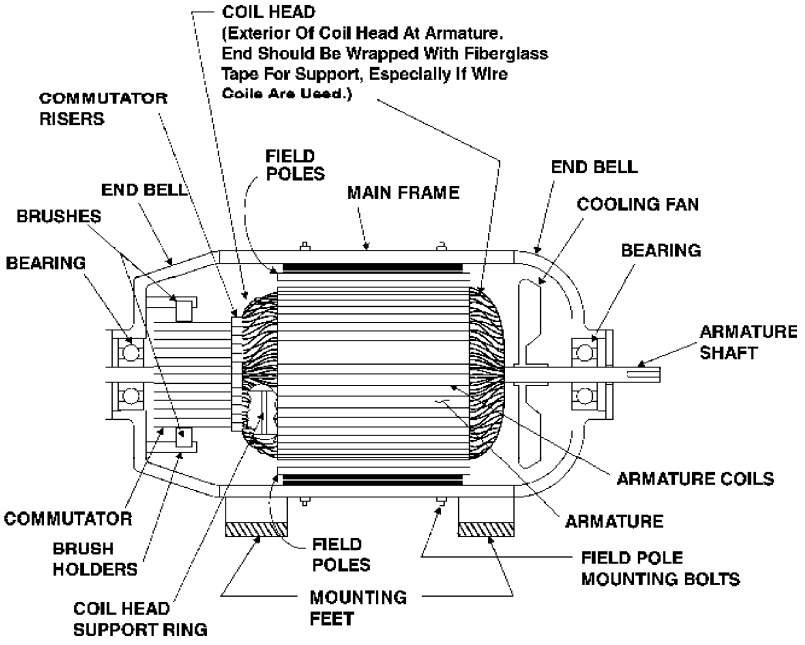

FIGURE 6.12Y
GENERAL DC MOTOR CONSTRUCTION

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Figures 6.12Y and 6.12Z picture the general construction of a DC motor as well as a 6 SCR (full-wave rectified) control circuit. The **Diagnostic Chart** (Figure 6.0) shows the many problems detectable through motor vibration signatures including broken armature windings, loose connectors, loose or blown fuses, shorted control cards and electrical fluting damage due to passage of electrical current through rolling element bearings. Each of these conditions is discussed in detail in higher level seminar texts.

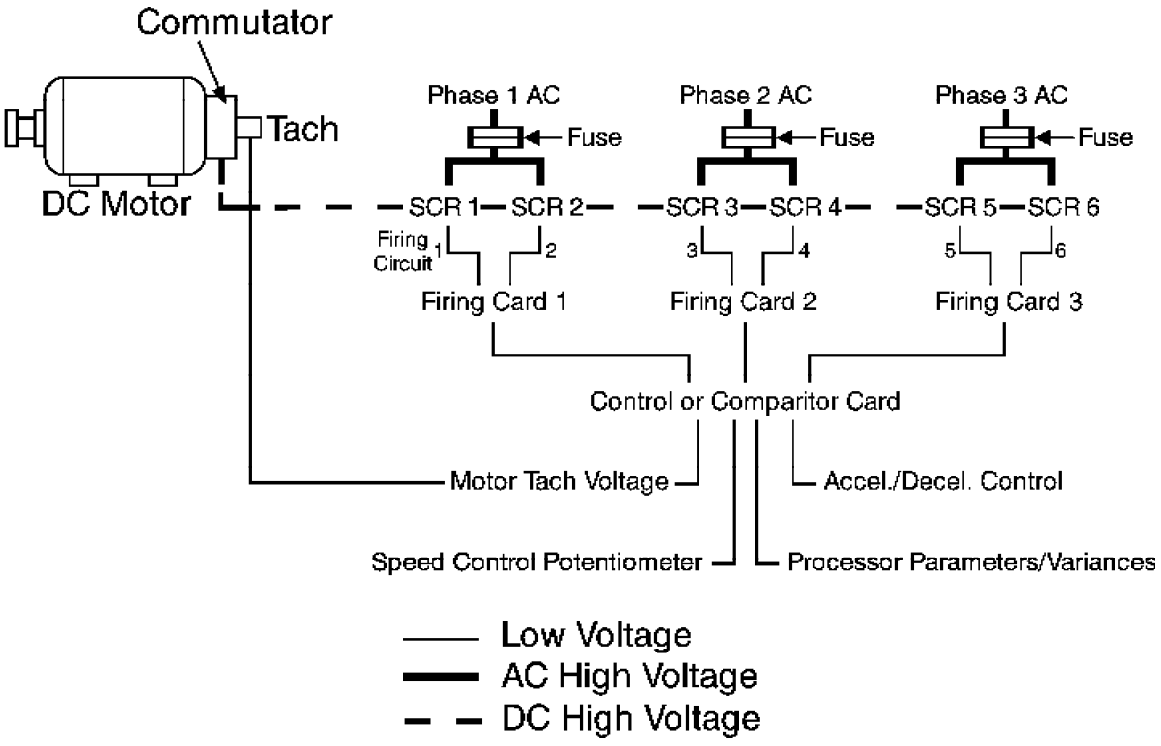

FIGURE 6.12Z
TYPICAL FULL-WAVE RECTIFIED, 6 SCR DC MOTOR CIRCUIT WITH 2 SCR'S PER FIRING CARD

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Figure 6.12AA illustrates a typical spectrum from a DC motor having a control problem due to a faulty SCR, shorted control card, loose connections and/or blown fuse. The problem is characterized by the pronounced appearance of multiple harmonics of the electrical line frequency (F_L) (none of which should be present in a good running DC motor with properly functioning controls). Only the inherent pulsing of the SCR's at 10,800 CPM (for a 3 SCR circuit) or 21,600 CPM (for a 6 SCR circuit) should be present, but at amplitudes typically below about .100 in/sec pk. Figure 6.12BB is a spectrum from a 6 SCR powered DC motor with a blown fuse and a shorted control card (universal summing amplifier card). Note the presence of F_L, $2F_L$, $3F_L$, $4F_L$, and $5F_L$. These F_L harmonics should not be present in a properly operating 6 SCR circuit. Figure 6.12CC was collected after repairs were made and shows no F_L harmonics were present after corrective actions were taken.

Again, a much more complete paper has been written concentrating not only on how each of the problems listed in the Diagnostic Chart (Figure 6.0) can be detected using vibration analysis, but also much information is provided on the construction of DC motors and controls and how they operate. This comprehensive DC motor paper (including real-world examples) is included within a higher level (Level III) seminar text, since it will require the analyst to have higher expertise in several related disciplines.

6.128 Torque Pulse Problems:

Electric motors inherently have vibration due to "torque pulses" created as the rotating magnetic field energizes the stator poles. Normally, this vibration is quite low and not a problem. The torque pulses themselves will occur at 2X line frequency (7200 CPM for a 60 Hz line frequency). This occurs since each motor pole is essentially energized 2X for each cycle of AC current.

This vibration problem is extremely rare except in those cases where exceptionally low vibration levels are mandated as in the case of machine tools, or if these torque pulses should happen to excite a natural frequency in the machine or structure located at or near twice line frequency. Torque pulses also can excite loose rotor bars and loose stator windings at frequencies of 2X, 3X and even 4X torque pulse frequency (or 14,400; 21,600; and 28,800 CPM).

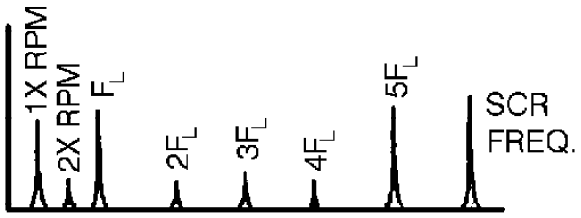

FIGURE 6.12AA
FAULTY SCR, SHORTED CONTROL CARD, LOOSE CONNECTIONS AND/OR BLOWN FUSE

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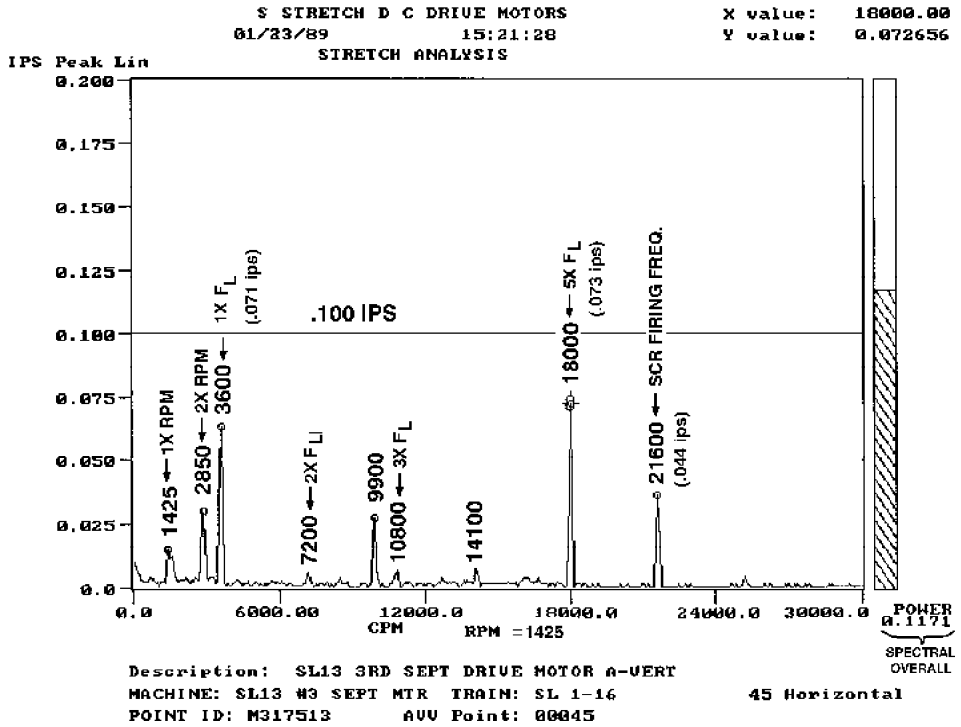

FIGURE 6.12BB
DC MOTOR SPECTRUM WITH A BLOWN FUSE AND SHORTED UNIVERSAL SUMMING CARD IN A 6 SCR CONTROL CIRCUIT

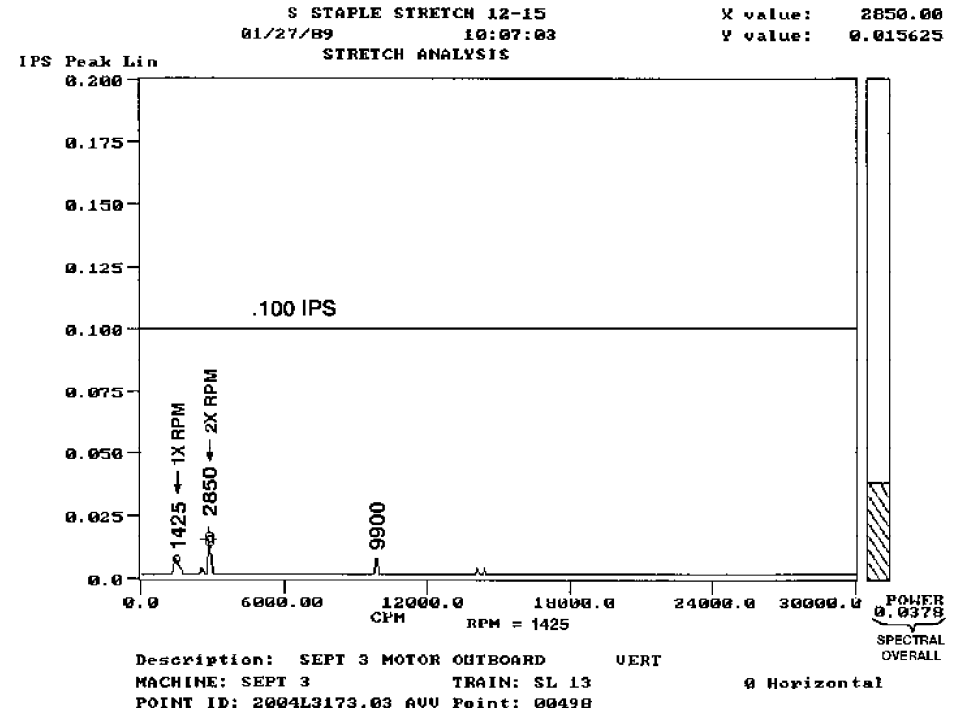

FIGURE 6.12CC
DC MOTOR SPECTRUM AFTER REPAIRS COMPLETED TO THE 6 SCR CONTROL CIRCUIT

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6.13 BELT DRIVE PROBLEMS

There are a whole variety of belt-driven machines ranging from those with inherently high vibration such as reciprocating air compressors to those requiring very low levels such as machine tool spindles. However, if the proper precautions are taken, the great majority of such machines should have low vibration. Probably 3 of the greatest factors affecting how much vibration a belt-driven machine will experience are:

- Alignment of Sheaves
- Sheave Concentricity
- Sheave Construction and Attachment Method

If each of these factors are carefully considered, there is no reason to have inherently high vibration in belt-driven machines.

Before discussing belt drive problems detectable by vibration analysis in particular, general statements concerning belt drives should be made:

1. It is best to take radial measurements in line with belt direction as shown in Figure 6.13A.

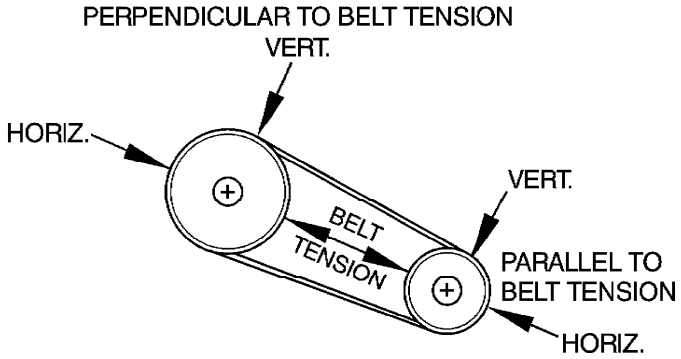

FIGURE 6.13A
PROPER MEASUREMENTS ON A BELT DRIVE

2. Adjustable V-belt sheaves create undue vibration and premature belt and sheave deterioration. These devices have inherent vibration problems since it is not possible to keep sheave faces parallel with one another which allows belts to ride up and down in the grooves with each revolution. As a result, this creates belt tension variation which generates high vibration and accelerates belt and sheave wear.

3. Another critical factor in belt drives is the amount of sheave eccentricity (i.e., runout). Unfortunately, sheaves which are purchased for general utility machinery almost always have inherently high eccentricity much greater than that of other components which are used in general rotating machinery. As a result, as soon as these are assembled to the machine, they themselves can generate high vibration and even cause noticeable variations in belt length and tension with each revolution. It is up to industry to demand tighter concentricity tolerance sheaves and to enforce these specifications right at machine acceptance.

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4. V-belt drives are often blamed as the source of high vibration when, in fact, they are simply reacting to other problems such as unbalance, misalignment, mechanical looseness, etc. When these other problems are present, they can cause high vibration in the belts which themselves are not the source of the problem.

Following below are discussions on many of the belt drive problems which can be detected by vibration analysis:

6.131 Worn, Loose or Mismatched Belts:

A typical spectrum indicating worn belts is given on page 4 of the **Illustrated Vibration Diagnostic Chart**. Note the belt frequency harmonics, all of which happen to be below both the driver and driven RPM in this example. The belt frequency (or belt RPM) is calculated as follows:

$$\text{Belt Freq.} = \frac{(3.142)(\text{Pulley RPM})(\text{Pulley Pitch Dia.})}{\text{Belt Length}}$$

When using the above equation, it is important to enter both the RPM and pitch diameter of the same pulley. It does not matter which pulley is used as long as the variables both come from the same pulley. Note that in all cases the belt RPM will be less than either the driver or driven RPM. However, belt frequency harmonics often will be higher than one or both of these. Worn, loose or mismatched belts display the following characteristics:

1. Worn belt defects detectable by vibration analysis include cracks, broken-off pieces of belt, hard and soft spots, lumps on belt faces, and also a crooked belt which has taken a set deformed shape during packing and storage.

2. When the problem is a worn belt(s), they will normally generate 3 to 4 multiples of belt frequency. Often, the 2X belt frequency peak may be dominant; in other cases, the fundamental belt frequency peak itself may not even show. In addition, worn belts sometimes lift the baseline of the spectrum throughout the subsynchronous frequency region, and just beyond driver and driven speeds as demonstrated in Figure 6.13B. In each case, worn belts normally cause unsteady amplitudes that sometimes pulsate with either the driver or driven RPM if any of these harmonics are close to either the driver or driven speed.

3. Belt defects usually show higher amplitudes in the direction parallel to belt tension. To get an idea how much vibration is caused by the belt defects themselves, compare amplitudes for the belt RPM frequencies themselves in a direction parallel with belt tension versus that in a direction perpendicular to belt tension.

4. Other belt specific problems which show up at belt RPM harmonics include belt width variations which cause the belts to ride up and down pulley grooves, creating vibration due to belt tension variations.

5. A loose cog belt is indicated by high vibration at the #cogs X RPM, and/or high vibration at the cog belt frequency itself which is calculated using the above equation.

6. Multiple V-belt drives can generate high vibration in the axial direction if they are unequally tensioned. This can result in excessive thrust bearing wear. These problems are sometimes solved by replacing several individual belts with one multi-belt which has been molded into one piece. If these are used, they place even greater importance on carefully aligning the sheaves.

7. Worn, loose or mismatched belts normally generate highest vibration in the radial direction, particularly in line with belt tension.

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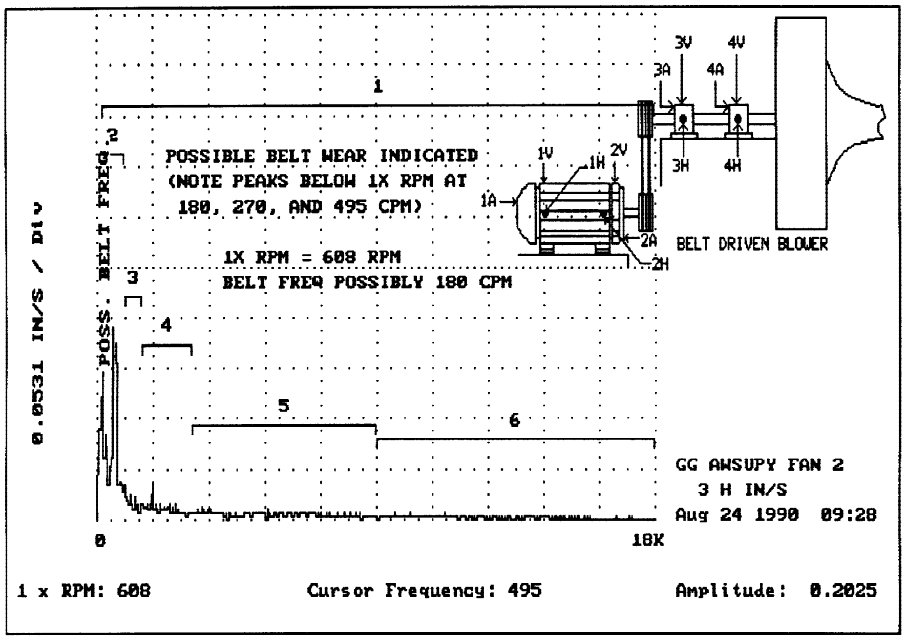

2% x 0.500 = 0.0100 1 x RPM: 608.00

Machine	Pos	Dir	Units	Date
GG AWSUPY FAN 2	3	H	IN/S	Aug 24 1990 09:28

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
180	0.296	0.1550	2520	4.145	0.0175
270	0.444	0.0900	2610	4.293	0.0100
495	0.814	0.2025	3240	5.329	0.0100
585	0.962	0.1850 = 1X RPM	3375	5.551	0.0125
765	1.258	0.0400	3600	5.921	0.0100
900	1.480	0.0375	5490	9.030	0.0100
1080	1.776	0.0250			
1215	1.998	0.0225			
1305	2.146	0.0150			
1440	2.368	0.0175			
1620	2.664	0.0175			
1800	2.961	0.0400			
1890	3.109	0.0175			
2025	3.331	0.0150			
2160	3.553	0.0125			
2340	3.849	0.0175			
2430	3.997	0.0125			

FIGURE 6.13B

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6.132 Belt/Sheave Misalignment:

Probably one of the greatest sources of belt drive vibration is misalignment of driver and driven sheaves. It is often amazing to see how much vibration can be reduced simply by employing a chalk line between sheaves. Misaligned sheaves display the following characteristics:

1. Axial vibration is most always generated by sheave misalignment which can highly accelerate the rate of wear of thrust bearings.

2. Misaligned sheaves produce high vibration at 1X RPM, predominately in the axial direction. Dominant vibration is quite often at driver RPM, but occasionally at driven RPM. The ratio of amplitudes of driver to driven RPM depends exactly on where the data is taken as well as on relative mass and frame stiffness.

3. Often with sheave misalignment, the highest axial vibration on the motor will be at fan RPM, while the highest axial when measuring on the fan will be at motor RPM. However, this is not always the case.

6.133 Eccentric Sheaves:

Eccentric sheaves are one of the greatest contributors to high vibration in belt-driven machines today, often due to a lack of emphasis on specifying good concentricity in purchase specs. Sheave eccentricity displays the following characteristics:

1. Eccentric and/or unbalanced sheaves cause high vibration at 1X RPM of the eccentric sheave.

2. Highest amplitude is normally in line with the belts and should show up on both the driver and driven sheaves.

3. Unlike unbalance, reaction forces caused by an eccentric pulley are not equally applied throughout the entire 360° rotation of the pulley. The force instead is concentrated in the direction of belt tension along a line passing through the centerline of the 2 shafts. As a result, this highly directional vibration will show comparative horizontal and vertical phase readings which either are identical or 180° opposite one another, depending on which side of the bearing the transducer is located for the measurement. In any case, both phase readings show that the bearing is moving in one line.

4. Since the forces are so directional in nature, the resultant vibration cannot be totally corrected by balancing via attaching washers to taperlock bolts. Even if balanced, the sheave eccentricity will still induce vibration in the belt due to belt length and tension variations, and will result in premature accelerated wear of belts and/or sheaves along with the driver and driven bearings.

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6.134 Belt Resonance:

Just like everything else in nature, belts themselves have a natural frequency which corresponds to the stiffness of the belt, the belt mass, and how much deflection is induced in the belt during operation. The natural frequency of a belt can be determined simply by pulling on the belt, releasing it and measuring the response. Belt resonance will display the following characteristics:

1. A spectrum indicating a belt resonance problem is shown on page 5 of the Table 6.0 **Illustrated Vibration Diagnostic Chart**. In this example, the belt natural frequency lies very close to 1X RPM of either the driver or driven machine. If this occurs, this will cause great flapping of the belt, particularly on the tension side at a frequency corresponding to the belt natural frequency.

2. Not only can a pulley speed excite a belt resonance, but also this can occur if a belt RPM harmonic should line up with its natural frequency.

3. Both the amplitude and phase of vibration at the belt resonant frequency will be unsteady.

4. The belt natural frequency itself can be changed either by altering the belt tension, belt length, sheave center distance, adding an idler pulley, etc..

6.135 Excessive Motor Vibration At Fan Speed Due to Motor Frame/Foundation Resonance:

Often, in the case of belt-driven machines, there will be excessive vibration on the motor. However, when a spectrum analysis is taken, it will show low vibration on the motor at motor speed, but high vibration on it at fan RPM. This can occur in either radial or axial directions. This will be evident by an excessive vibration at 1X RPM of the driven unit, particularly in one direction on the motor. Often, when this occurs, the problem source is excitation of a motor frame or foundation natural frequency by the incoming fan speed vibration. This can be confirmed by simple impulse natural frequency tests explained in Section 6.05. In this case, the solution will normally be stiffening of the frame or foundation by addition of bracing or addition of concrete on the base if it is resonant.

6.136 Loose Pulley or Fan Hub:

Excessive vibration sometimes occurs in belt-driven machines due to looseness either of a pulley or a fan hub itself. This is evidenced by each of the following characteristics:

1. There will be excessive vibration particularly at 1X RPM, but also at several running speeds harmonics.

2. One of the best indicators will be unstable phase. If operating properly, the phase in horizontal, vertical and axial directions should be steady. If a fan hub held on by set screws is loose on the shaft, it may show a difference in both phase and amplitude each time the unit is started up. In these cases, balancing would be only a temporary solution. It may help for a “couple of hours”, but later when the fan hub or pulley rotates slightly on the shaft, it will upset the whole balance possibly dramatically changing the phase and amplitude. This same thing can occur if either the pulley or fan hub has a taper fit, and is not properly pulled up and fastened to the shaft.

3. The solution to each of these cases is ensuring that all rotors attached to the shaft are securely fastened, and properly oriented in the case of a taper fit.

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6.14 BEAT VIBRATION

Beat vibration can occur either when 2 frequencies lie close to one another, or a single frequency is continually changing in amplitude and frequency. More often than not "beat vibration" occurs when 2 or more steady-state vibration sources generate frequencies in close proximity to one another.

For example, if 2 fans are mounted on the same foundation and happen to operate at nearly identical speeds, they will generate a beat vibration which can both be felt and heard. For example, if one fan were operating at 900 RPM and the other at 950 RPM, they would generate a beat frequency of 50 RPM. When viewed in an analyzer with insufficient resolution to separate the 2 nearly identical frequencies, the analyst will see one peak pulsating up and down. For example, if his maximum frequency were set at 60,000 CPM (150 CPM resolution), the analyzer would display one peak at 900 CPM. If he could see this in "real-time", the peak would pulsate. If the analyst has "zoom capability", he can zoom in on this so called 900 CPM frequency and resolve it into the 2 individual peaks. Also, if the analyst had a low frequency seismic transducer, he could likewise detect the beat frequency itself down at 50 CPM.

Audible beat frequencies are common in the case of induction electric motors which have stator or rotor problems. In these cases, a running speed harmonic will often occur close to twice line frequency, again causing a beat. For example, if a 3580 CPM motor developed an eccentric rotor problem causing high levels at 7200 CPM, its 2nd running speed harmonic at 7160 CPM would beat with 2X line frequency (7200 CPM), and would likewise generate a beat frequency itself at 40 CPM.

Figure 6.14A demonstrates what happens in a beat frequency. Here each of 2 time waveforms that correspond to 2 individual frequencies are shown in the upper part of the figure. Note that the time waveforms are slightly different meaning that the frequencies are likewise slightly different. However, periodically both waveforms peak at the same time as they come into phase with one another. At this point, they effectively add to one another causing the highest amplitudes as seen in the beat frequency time waveform itself in the lower diagram in this figure. Referring back to the upper figure, note that at other times, the time waveforms of the 2 frequencies are 180° out of phase, effectively cancelling one another. Looking directly below this point on the lower time waveform, note that the amplitude here goes almost to 0. Therefore, this explains why one feels and hears pulsating noise and vibration. Note that the lower time waveform is in fact the beat frequency generated by the 2 frequencies. As the figure shows, this beat frequency (F_S) is in fact the difference frequency ($F_1 - F_2$).

Figure 6.14B illustrates beat vibration on a real machine. This spectrum was taken on a large gearbox whose output was to a long jackshaft. Note that the input to the gearbox from the drive motor was at 1477 RPM and that one of its outputs to the jackshaft was at 1395 RPM (a difference of only 82 CPM). When looking at a much wider frequency scale, it combined these 2 individual peaks into one and showed them to be pulsating greatly. Figure 6.14C shows the beat frequency itself which was the difference between the higher drive motor and lower jackshaft operating speeds.

Beat frequencies themselves are not necessarily a problem. However, if one desires to balance one of the machines, for example, he will find it very difficult, particularly if attempting to do so with a strobe light instrument. If a beat frequency is involved, often the strobe light image will continually rotate at the beat frequency itself. It becomes impossible to proceed with any type of balance since phase cannot be measured.

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In another scenario, 2 machines running alone may be within vibration severity criteria. However, when both are running, the resultant is that both machines are thrown above alarm limits. In this case, one solution is to change the speed of one or both units so that they will no longer beat. Another solution is to place vibration isolation materials under one or both machines to keep the vibration within each from affecting the other. In general, it is a good idea to keep significant forcing frequencies a minimum of approximately 10% away from one another in order to prevent the pulsating beat frequency vibration. In any case, beat frequencies are not normally a problem when the differences exceed 150 to 200 CPM.

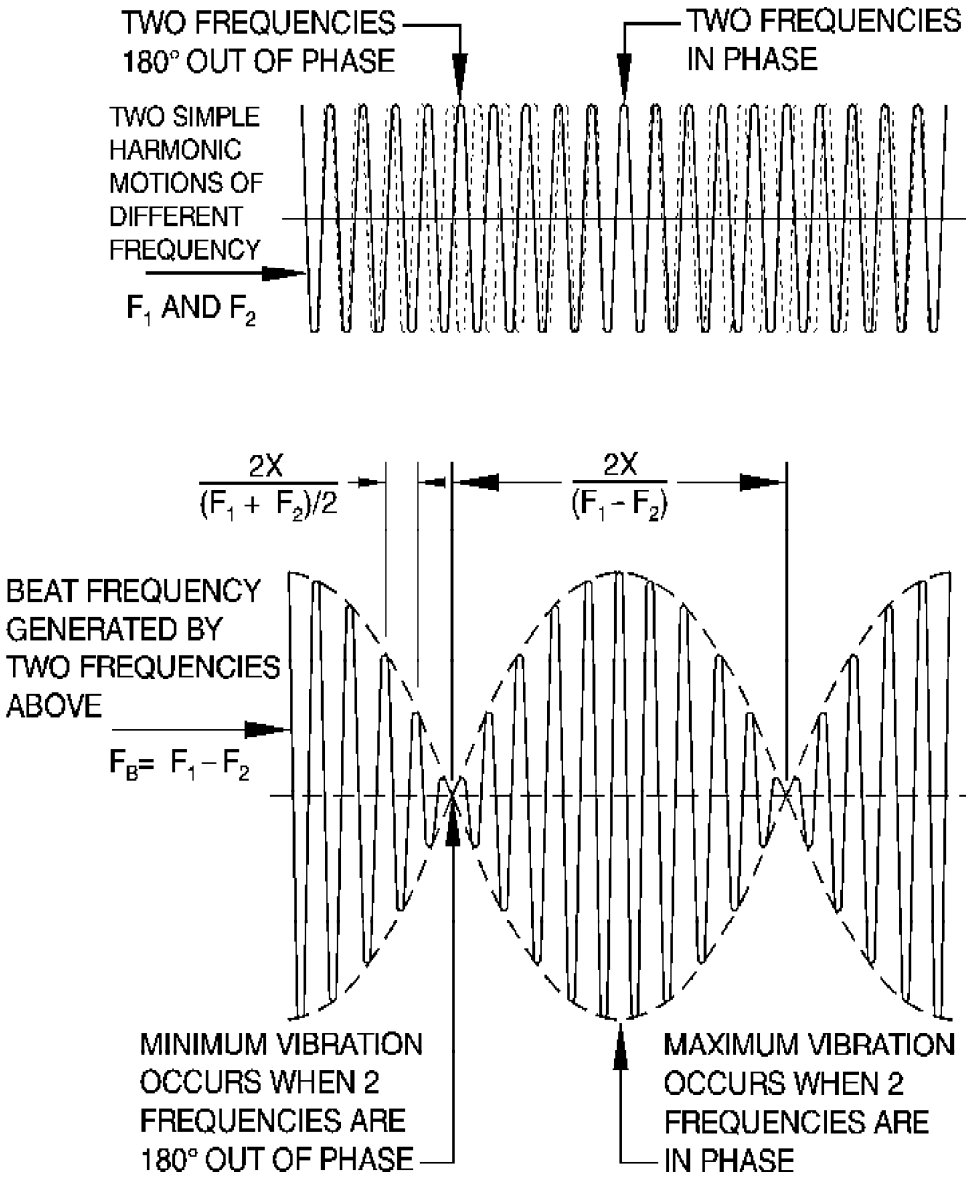
TWO FREQUENCIES
180° OUT OF PHASE
TWO FREQUENCIES
IN PHASE
TWO SIMPLE HARMONIC MOTIONS OF DIFFERENT FREQUENCY
F1 AND F2
2X
(F1 + F2)/2
2X
(F1 - F2)
BEAT FREQUENCY GENERATED BY TWO FREQUENCIES ABOVE
FB= F1 - F2
MINIMUM VIBRATION OCCURS WHEN 2 FREQUENCIES ARE 180° OUT OF PHASE
MAXIMUM VIBRATION OCCURS WHEN 2 FREQUENCIES ARE IN PHASE

FIGURE 6.14A
ILLUSTRATION OF A BEAT FREQUENCY

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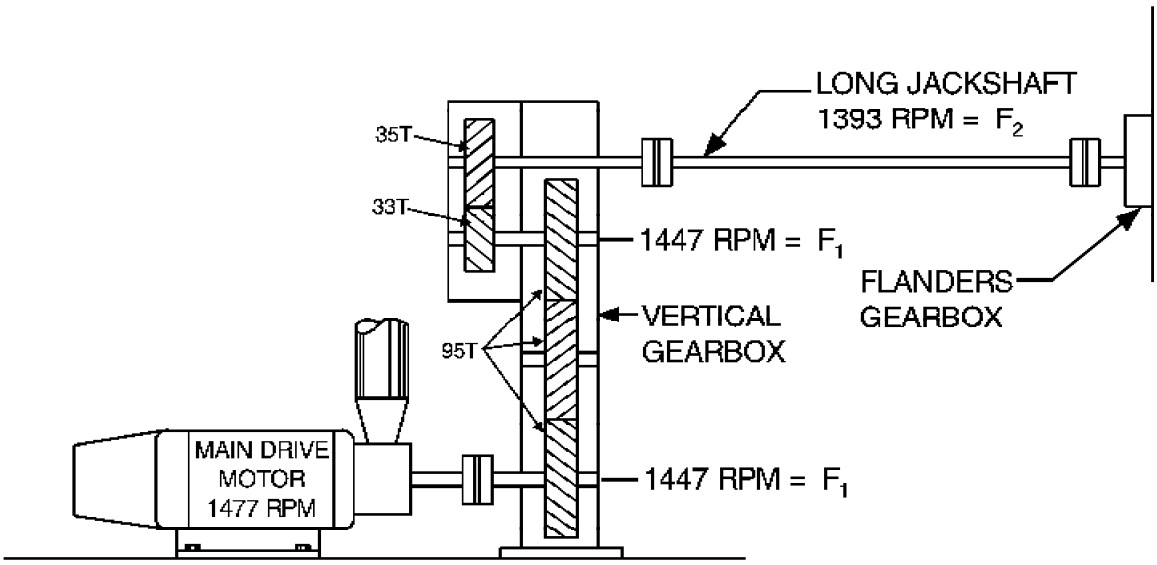

NOTE: BEAT VIBRATION FREQUENCY = MOTOR SPEED - JACKSHAFT SPEED

$$F_B = F_1 - F_2$$

$$F_B = 1477 \text{ RPM} - 1393 \text{ RPM}$$

BEAT VIBRA. FREQ. = F_B = 84 RPM

(SEE FIGURES 6.14C AND 6.14D)

FIGURE 6.14B
MAIN DRIVE FOR DRAW FRAME NO. 1
(NOTE: F_B, F_1 AND F_2 ARE PICTURED IN THEIR TIME WAVEFORMS ON FIGURE 6.14A)

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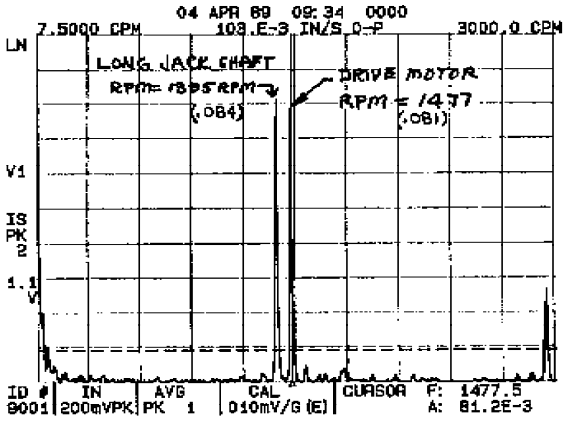


PEAK PICK ON TRACE 1

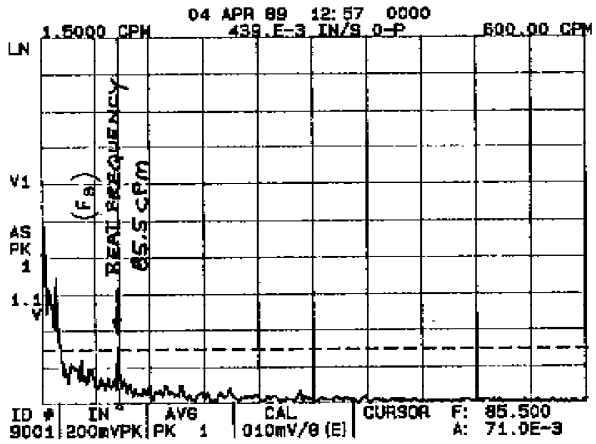
IN/S 0-P	CPM	
20.7E-3	37.500	
9.88E-3	60.000	
84.2E-3=.084	1394.9	← JACK SHAFT
81.4E-3=.081	1477.4	← MOTOR
27.7E-3	2955.0	

ΔRPM=82.5

DRAW FRAME 1 - HIGH SPEED

DRAW MACHINE NO. 1
VERTICAL ON FLOOR
OVERALL VIB. = .128"/sec

FIGURE 6.14C
PULSATING AMPLITUDES OF JACKSHAFT & MOTOR SPEED FREQUENCIES DUE TO BEAT VIBRATION
(AMPLITUDES PULSATED APPROX. ±50%, OR .04 UP TO .12 IN/SEC EACH)



PEAK PICK ON TRACE 1

IN/S 0-P	CPM	
214.E-3	3.0000	
140.E-3	7.5000	
152.E-3	16.500	
70.2E-3	85.500	←

DRAW MACHINE NO. 1
VERTICAL ON FLOOR

FIGURE 6.14D
GENERATION OF BEAT FREQUENCY ITSELF

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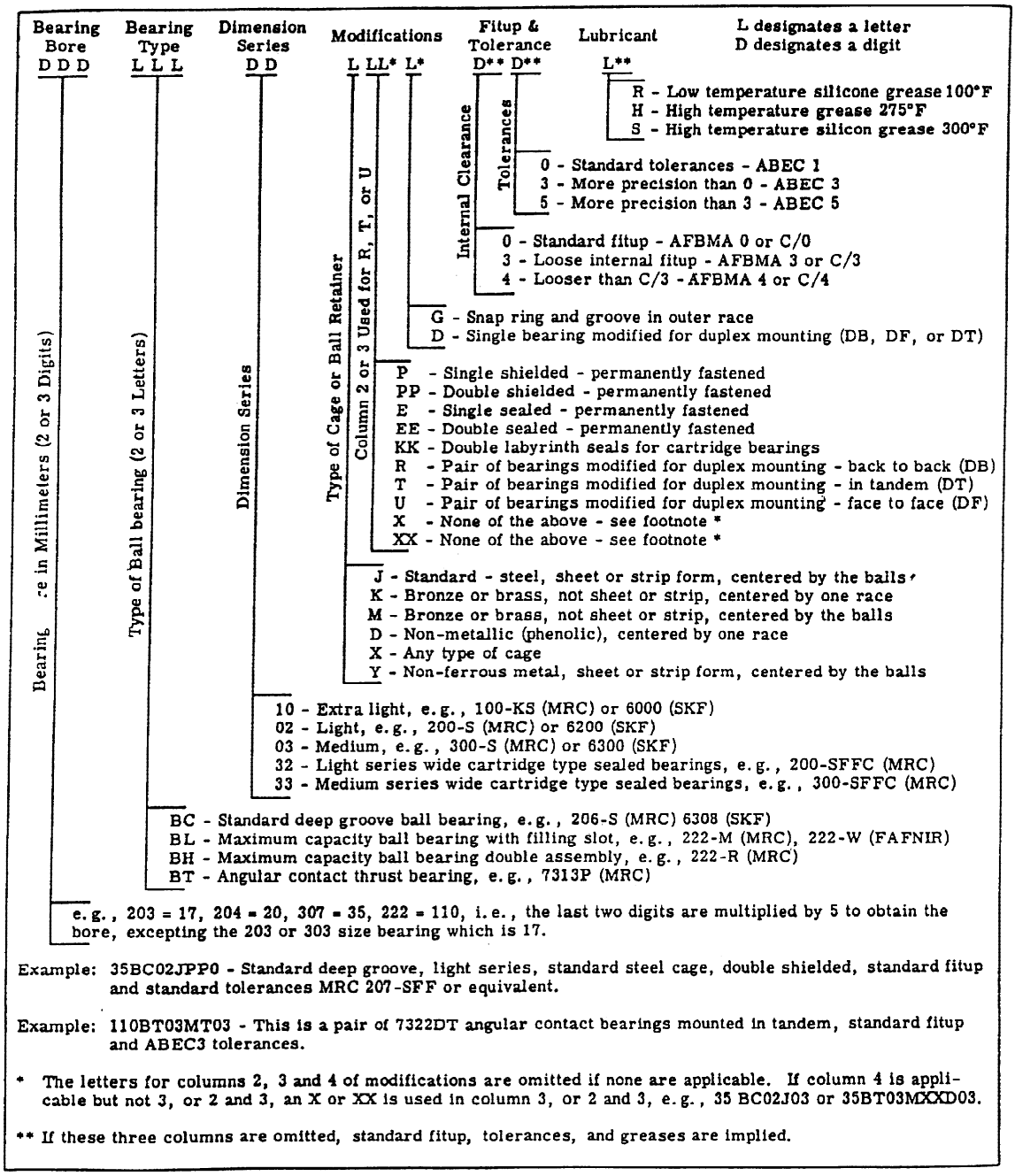
APPENDIX

AFBMA NOMENCLATURE — BALL BEARINGS


Bearing Bore D D D
Bearing Type L L L
Dimension Series D D
Modifications L LL* L*
Fitup & Tolerance D** D**
Lubricant L**
L designates a letter
D designates a digit
Bearing Bore in Millimeters (2 or 3 Digits)
Type of Ball bearing (2 or 3 Letters)
Dimension Series
Type of Cage or Ball Retainer
Column 2 or 3 Used for R, T, or U
Internal Clearance
Tolerances
R - Low temperature silicone grease 100°F
H - High temperature grease 275°F
S - High temperature silicon grease 300°F
0 - Standard tolerances - ABEC 1
3 - More precision than 0 - ABEC 3
5 - More precision than 3 - ABEC 5
0 - Standard fitup - AFBMA 0 or C/0
3 - Loose internal fitup - AFBMA 3 or C/3
4 - Looser than C/3 - AFBMA 4 or C/4
G - Snap ring and groove in outer race
D - Single bearing modified for duplex mounting (DB, DF, or DT)
P - Single shielded - permanently fastened
PP - Double shielded - permanently fastened
E - Single sealed - permanently fastened
EE - Double sealed - permanently fastened
KK - Double labyrinth seals for cartridge bearings
R - Pair of bearings modified for duplex mounting - back to back (DB)
T - Pair of bearings modified for duplex mounting - in tandem (DT)
U - Pair of bearings modified for duplex mounting - face to face (DF)
X - None of the above - see footnote *
XX - None of the above - see footnote *
J - Standard - steel, sheet or strip form, centered by the balls
K - Bronze or brass, not sheet or strip, centered by one race
M - Bronze or brass, not sheet or strip, centered by the balls
D - Non-metallic (phenolic), centered by one race
X - Any type of cage
Y - Non-ferrous metal, sheet or strip form, centered by the balls
10 - Extra light, e.g., 100-KS (MRC) or 6000 (SKF)
02 - Light, e.g., 200-S (MRC) or 6200 (SKF)
03 - Medium, e.g., 300-S (MRC) or 6300 (SKF)
32 - Light series wide cartridge type sealed bearings, e.g., 200-SFFC (MRC)
33 - Medium series wide cartridge type sealed bearings, e.g., 300-SFFC (MRC)
BC - Standard deep groove ball bearing, e.g., 206-S (MRC) 6308 (SKF)
BL - Maximum capacity ball bearing with filling slot, e.g., 222-M (MRC), 222-W (FAFNIR)
BH - Maximum capacity ball bearing double assembly, e.g., 222-R (MRC)
BT - Angular contact thrust bearing, e.g., 7313P (MRC)
e.g., 203 = 17, 204 = 20, 307 = 35, 222 = 110, i.e., the last two digits are multiplied by 5 to obtain the bore, excepting the 203 or 303 size bearing which is 17.

Example: 35BC02JPP0 - Standard deep groove, light series, standard steel cage, double shielded, standard fitup and standard tolerances MRC 207-SFF or equivalent.

Example: 110BT03MT03 - This is a pair of 7322DT angular contact bearings mounted in tandem, standard fitup and ABEC3 tolerances.

* The letters for columns 2, 3 and 4 of modifications are omitted if none are applicable. If column 4 is applicable but not 3, or 2 and 3, an X or XX is used in column 3, or 2 and 3, e.g., 35 BC02J03 or 35BT03MXXD03.

** If these three columns are omitted, standard fitup, tolerances, and greases are implied.

Table 2-10

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APPENDIX

AFBMA NOMENCLATURE — ROLLER BEARINGS

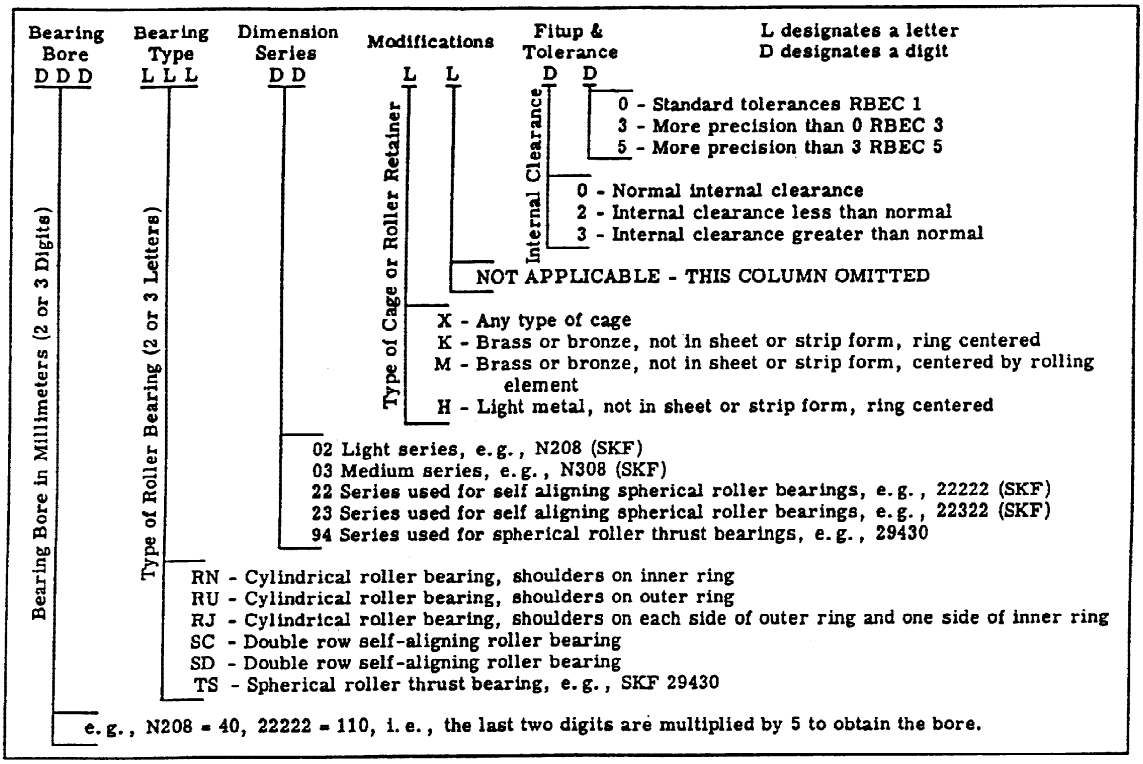
Bearing Bore
D D D
Bearing Type
L L L
Dimension Series
D D
Modifications
L L
Fitup & Tolerance
D D
L designates a letter
D designates a digit
Bearing Bore in Millimeters (2 or 3 Digits)
Type of Roller Bearing (2 or 3 Letters)
Type of Cage or Roller Retainer
Internal Clearance
0 - Standard tolerances RBEC 1
3 - More precision than 0 RBEC 3
5 - More precision than 3 RBEC 5
0 - Normal internal clearance
2 - Internal clearance less than normal
3 - Internal clearance greater than normal
NOT APPLICABLE - THIS COLUMN OMITTED
X - Any type of cage
K - Brass or bronze, not in sheet or strip form, ring centered
M - Brass or bronze, not in sheet or strip form, centered by rolling element
H - Light metal, not in sheet or strip form, ring centered
02 Light series, e.g., N208 (SKF)
03 Medium series, e.g., N308 (SKF)
22 Series used for self aligning spherical roller bearings, e.g., 22222 (SKF)
23 Series used for self aligning spherical roller bearings, e.g., 22322 (SKF)
94 Series used for spherical roller thrust bearings, e.g., 29430
RN - Cylindrical roller bearing, shoulders on inner ring
RU - Cylindrical roller bearing, shoulders on outer ring
RJ - Cylindrical roller bearing, shoulders on each side of outer ring and one side of inner ring
SC - Double row self-aligning roller bearing
SD - Double row self-aligning roller bearing
TS - Spherical roller thrust bearing, e.g., SKF 29430
e.g., N208 = 40, 22222 = 110, i.e., the last two digits are multiplied by 5 to obtain the bore.

Table 2-11

RADIAL BALL BEARINGS INTERCHANGEABILITY CHART

	SKF	MRC	FAFNIR	NORMA HOFFMAN	HOOVER	NEW DEPARTURE
Deep groove, radial, light series, open type	6200	200S	200K	200	200	3200
Deep groove, radial, medium series, open type	6300	300S	300K	300	300	3300
Deep groove, radial, light series, single shielded	6200Z	200SF	200KD	200P	7200	7500
Deep groove, radial, medium series, single shielded	6300Z	300SF	300KD	300P	7300	7600
Deep groove, radial, light series, double shielded	62002Z	200SFF	200KDD	200PP	77200	77500
Deep groove, radial, medium series, double shielded	63002Z	300SFF	300KDD	300PP	77300	77600

Table 2-12

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CHAPTER 7

PROVEN METHOD FOR SPECIFYING BOTH 6 SPECTRAL ALARM BANDS AS WELL AS NARROWBAND ALARM ENVELOPES USING TODAY'S PREDICTIVE MAINTENANCE SOFTWARE SYSTEMS (3rd Edition)

By: James E. Berry, P.E.
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James E. Berry has 21 years mechanical engineering experience including 19 years specializing in Machinery Vibration Diagnostics, setup and implementation of Predictive Maintenance Programs, Modal Analysis and Stress Analysis. Mr. Berry received both his Bachelor of Science (1973) and Masters Degrees (1974) in Mechanical Engineering from North Carolina State University. He is also a registered professional engineer and is an active member of both the Vibration Institute and ASA. He has published several vibration analysis articles in technical journals such as Sound and Vibration Magazine, given technical papers and seminars to the Vibration Institute, and has given presentations to several engineering societies including ASME, AIPE, ASA, and ASTME. Serving as a Consulting Engineer, he has performed vibration analysis on a wide variety of both process and utility machinery for a diverse group of clientele served by Technical Associates.

7.0 ABSTRACT

Although there is much literature available on how to diagnose machine problems using vibration analysis, there is little material available today on how to specify effective spectral alarm bands on various types of machinery. These spectral alarm bands are now offered within the software of several vendors serving the field of predictive maintenance, and thanks to these vendors, provide the potential of detecting numerous machine problems that might otherwise go unnoticed. In the detection and analysis process, if these spectral bands are utilized, they can save the user thousands of dollars in maintenance expenditures and make significant impact on improving plant profitability. First, one needs to know that his machine has a problem. Then, he must take steps to diagnose both the source of the problem and determine its severity. The purpose of this paper is to provide a documented technique on how to specify peak velocity spectral alarm levels and frequency bands for measurements taken on the housings of general process and utility machinery. If properly specified for the specific machine type, drive configuration, bearing type and operating speed, these spectral alarm bands will notify the user that he does have a problem without generating a series of false alarm "emergencies". The techniques included are not intended, and will not apply to all machine types under all operating conditions. However, they have been successfully applied to a diverse array of machinery ranging from common pumps and blowers to refrigeration chillers, hammer mills, precision machine tool spindles, high-speed centrifugal air compressors, moderate speed rolling mill drives, etc.

This paper now applies not only to those software systems which allow 6 spectral alarm bands ("power bands"), but also to those which allow generation of narrowband alarms which place a threshold envelope around individual frequencies which are caused by specific sources within the machine ("threshold band alarms").

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7.1 INTRODUCTION TO SPECIFYING SPECTRAL ALARM BANDS & FREQUENCY RANGES

Properly specified spectral alarm bands are probably one of the most critical weapons in the predictive maintenance arsenal today for detecting potentially serious problems which develop in machinery. However, although thousands of data collectors are now in place throughout the world with software having the capability of comparing each new FFT spectrum captured to these user-defined spectral alarm bands, surveys have shown that very few users have sufficient experience to know how to properly and effectively set up these bands in their computers. In fact, a large percentage of plants abound that do not even use spectral alarm band capabilities though they are offered in their software. Instead, they depend on trending of their overall levels to warn them of impending machine problems. On the other hand, many of those who themselves have made concerted efforts to specify and use these bands often complain that they do not feel very comfortable with the bands they have specified; and do not have the time required to learn how to specify one set for one machine type and an entirely different set for another, depending on how the machine is configured (bearing type, operating speed, drive configuration, etc.). Many users at these plants are hard pressed just to determine what overall alarm vibration levels should be specified for these machines, much less have the time to research how to specify individual band frequency ranges and alarm levels. Many spend hundreds of man-hours just trying to specify the optimum overall, and even then, doubt how meaningful these are.

As we now know from spectrum analysis, if it were possible to obtain so-called “perfect overall vibration specifications”, potentially serious problems can still develop within machines, and yet, can have negligible effect on the level of overall vibration. However, these same problems would be noticeable in the FFT spectrum. But, if no spectral alarm system were in place to even detect the presence of a problem, the user would be unaware of its existence until possibly considerable damage had been done, not only to one component, but possibly to several other components in this machine as well.

Such problems as deteriorating bearings, gears, electrical problems, etc. may not make themselves known for some time to those who depend only on overall levels to detect problems. For example, a bearing defect frequency might grow by a factor of 4X from .03 to .12 in/sec and cause almost no change in the overall if amplitudes at 1X and 2X RPM were, say .35 in/sec and .20 in/sec (in this case, the 4X increase by the bearing frequency would only increase the overall from approximately .40 to .42 in/sec). Even though there would be a definite increase in bearing defect vibration, the overall alone would simply not be sufficiently sensitive to show there was any real change.

Therefore, the express purpose of this paper is to offer a procedure for specifying meaningful spectral alarm bands for a variety of machine types and configurations. This method has been in use by our company for several years as we have set up and carried out complete predictive maintenance programs for a large number of clients in a diverse array of industries. This technique is summarized in Table III.

This technique is not meant (or implied) to specify concrete spectral alarm bands that cannot be altered, no matter what the unique spectral characteristics of a given machine family. Instead, it is meant to provide a firm starting point to:

(1) Allow a plant having no prior experience or machine vibration history to initially set up effective spectral alarm bands for hundreds of machines in his plant prior to making baseline measurements;

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(2) Allow a more experienced plant to set up spectral alarm bands for the first time, even though it might have captured data on large numbers of machines for several years, but has never set up the bands due to a lack of understanding of how to properly do so;

(3) Allow the plant which in fact has installed spectral alarm bands to objectively compare them to other setups, and to evaluate how effective their current bands are.

Please keep in mind that after several samples of data have been acquired, the user should carefully review how each setup for each grouping of machines is working (assuming it is possible to place all his machines into specific groupings or families). Included in the paper will not only be how to specify spectral alarm bands, but also suggestions on how to evaluate their effectiveness and refine them as well.

7.11 TWO TYPES OF SPECTRAL ALARM BANDS

Importantly, note that two different types of spectral alarm bands are used by several different predictive maintenance software versions - 1) Absolute Threshold, and 2) so-called "**Power Band**" type. It is important that the user know which type is employed by his predictive maintenance software system and take this into account.

Absolute threshold systems enable users to specify the maximum allowable amplitude of any single peak within each band. If any peak equals or exceeds this threshold, it will cause the band to go into alarm.

On the other hand, power band systems calculate the total energy (or "power") within each band generated by all the peaks within that band. The total power within each band is calculated using the same equation as that used to determine the overall level of an entire spectrum as per the following formula:

$$OA = \frac{\sqrt{\sum_{i=1}^{n} A_i^2}}{\sqrt{N_{BF}}} = \frac{\sqrt{A_1^2 + A_2^2 + A_3^2 + ... + A_N^2}}{\sqrt{1.5}}$$

EQUATION (1)

Where:

OA = Overall Level of Spectrum or Band
n = Number of FFT Lines of Resolution
A_1 = Amplitude of Each of the FFT Lines
N_{BF} = Noise Bandwidth for Window Chosen
1.5 for Hanning Window

Which means $OA = .8165 \sqrt{\sum_{i=1}^{n} A_i^2}$

Note that it is not necessary for any individual peak to equal or exceed the band alarm in order to exceed the alarm for that band. That is, if a band were specified to extend from .50X F_{MAX} through F_{MAX} using a 400 line FFT, Equation (1) would be used to calculate the power from the 200th line through the 400th line of the spectrum. For example, for a power band having a .20 in/sec specified alarm, the energy of only 2 peaks within this band having amplitudes of .175 and .185 in/sec, respectively, would exceed this alarm.

Importantly, the spectral alarm band amplitudes specified in Table III assume the power band type. The same frequency ranges would apply for either alarm band system. However, if one had absolute threshold bands, he should lower alarm levels specified in Table III somewhat - probably on the order of 20% to 30%, particularly for those bands having wider frequency ranges.

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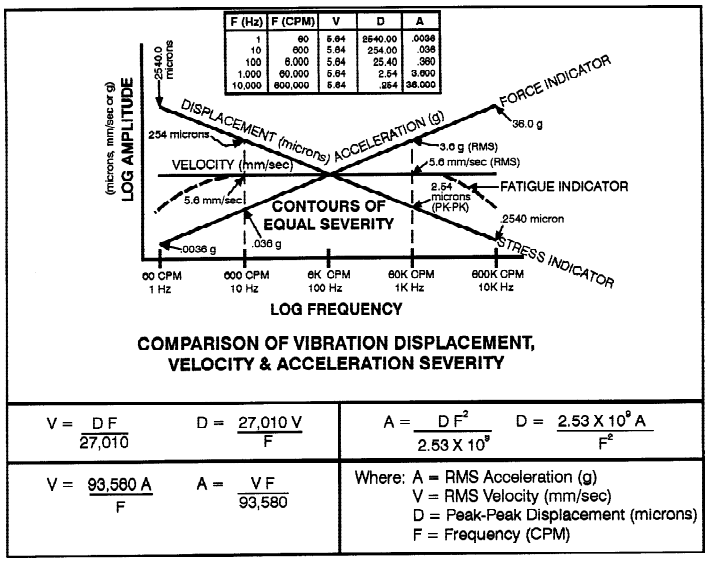
7.12 WHICH VIBRATION PARAMETER TO USE IN SPECTRAL ALARM BANDS - DISPLACEMENT, VELOCITY OR ACCELERATION?

Three important items must be understood when setting up spectral alarm bands for machines. First, one must know what forcing frequencies will be generated by such things as rolling element bearing wear, sleeve bearing wear, gear problems, electrical problems, unbalance, misalignment, etc. Secondly, however, he must then know which vibration parameter (displacement, velocity or acceleration) will best detect those problems he will see on his particular machines. Thirdly, he must know how many FFT lines must be used to even show the presence of such problems.


F (Hz) | F (CPM) | V | D | A
1 | 60 | 5.64 | 2540.00 | .0036
10 | 600 | 5.64 | 254.00 | .036
100 | 6,000 | 5.64 | 25.40 | .360
1,000 | 60,000 | 5.64 | 2.54 | 3.600
10,000 | 600,000 | 5.64 | .254 | 36.000
LOG AMPLITUDE
(microns, mm/sec or g)
2540.0 microns
DISPLACEMENT (microns)
ACCELERATION (g)
VELOCITY (mm/sec)
254 microns
5.6 mm/sec
FORCE INDICATOR
36.0 g
3.6 g (RMS)
5.6 mm/sec (RMS)
FATIGUE INDICATOR
2.54 microns (PK-PK)
.2540 micron
CONTOURS OF EQUAL SEVERITY
.036 g
.0036 g
STRESS INDICATOR
60 CPM 1 Hz
600 CPM 10 Hz
6K CPM 100 Hz
60K CPM 1K Hz
600K CPM 10K Hz
LOG FREQUENCY

COMPARISON OF VIBRATION DISPLACEMENT, VELOCITY & ACCELERATION SEVERITY

$V = \frac{D\,F}{27{,}010}$ $\quad D = \frac{27{,}010\,V}{F}$	$A = \frac{D\,F^2}{2.53 \times 10^9}$ $\quad D = \frac{2.53 \times 10^9\,A}{F^2}$
$V = \frac{93{,}560\,A}{F}$ $\quad A = \frac{V\,F}{93{,}580}$	Where: A = RMS Acceleration (g) V = RMS Velocity (mm/sec) D = Peak-Peak Displacement (microns) F = Frequency (CPM)

FIGURE 1
CONTOURS OF EQUAL SEVERITY AND CONVERSION FORMULAS

For example, some users still acquire displacement spectra for most all of their machinery because "this is the way it has been done for many years". Figure 1 shows that while displacement does a good job on (and is the most sensitive parameter to) low frequency measurements predominantly below 600 CPM, it does not adequately detect problems which are higher frequency in nature such as rolling element bearing and gear wear. For example, assume a belt-driven blower with a nominal 3600 RPM motor, having a serious bearing defect frequency amplitude of .30 in/sec (peak) at 60,000 CPM, as well as a .30 in/sec amplitude at running speed (1X RPM). If peak-peak displacement were used to evaluate this machine, the equations in Figure 1 show that the .30 in/sec level at 60,000 CPM would correspond to a deceptively low amplitude of only .095 mil at 60,000 CPM; while the running speed vibration would be approximately 1.59 mils at 3600 CPM. Since normal alarms for this machine would be set at approximately 2 mils, the 3600 RPM running speed vibration would readily be visible in a linear amplitude spectrum, while that at the 60,000 CPM bearing frequency would not. This would give one a false sense of security concerning his bearing health. He may not even notice the presence of the bearing defect peak, and therefore, would not even attempt to determine its source. On the other hand, velocity spectra would clearly display the bearing frequency peak and would show it at just as high in amplitude as that at operating speed (but in reality, of much greater problem severity than that at 1X RPM).

For another example, even though displacement would be the best indicator of unbalance or misalignment on a machine running only 300 RPM, if one's primary interest were rolling element bearing condition, velocity would again be the best parameter to employ. This is commonly the case with paper machines and other large, low-speed machinery.

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On the other hand, velocity spectra have their limitations as well. For example, assume a common centrifugal air compressor running at 3580 RPM and having a 344 tooth bullgear. This machine would have a fundamental gear mesh frequency (GMF) of approximately 1,231,500 CPM (20,525 Hz). Experience with these machines proves, that not only must one evaluate amplitude at the fundamental gear mesh frequency, but at least at the second and third GMF harmonic as well. A good conditioned, well aligned set of gears will normally have a level of approximately 6 g (peak) at 2X GMF of approximately 2,463,000 CPM (41,050 Hz). If these gears were to develop definite problems increasing the amplitude 10 times higher to 60 g at 2X GMF, this would correspond to a peak velocity of only .089 in/sec (and an even more deceptive peak-peak displacement of only .00070 mil).

Therefore, the best indicator of problems which generate forcing frequencies in these high frequency regions, particularly above 300,000 CPM, is acceleration.

However, velocity spectra will prove to be the best indicator of a large majority of problems likely on about 80% to 90% of rotating machinery. Therefore, the emphasis of this article will be on specification of peak velocity spectral alarm bands (again, one can easily convert peak velocity overall amplitudes shown in Table II and spectral alarm band amplitudes shown in Table III to RMS simply by multiplying levels by a factor of .707). In fact, the great majority of instruments available actually make RMS measurements, and "convert" them to so-called peak velocity (or peak acceleration) by simply multiplying the amplitude measurement for each frequency by 1.414 ($\sqrt{2}$).

Similar bands can be specified for either displacement or acceleration for those machines requiring these parameters. Then, it will be important to take into account how displacement and acceleration vary with the frequency (see formulas in Figure 1). For example, assume that when using velocity, one were going to set the alarm level for 1X RPM at .30 in/sec, while setting the alarms at 2X and 3X RPM at one half this amount, or .15 in/sec. Since displacement varies directly with velocity, but inversely with frequency (see Figure 1), if he sets the 1X RPM displacement alarm at 6 mils, he will need to set the 2X RPM alarm level at only 1/4 that at 1X RPM (1.5 mil) and the 3X RPM alarm at only 1/6 that at 1X RPM (1 mil).

On the other hand, given the same machine, if one wished to convert from velocity to acceleration bands, he would need to make the relationship of one harmonic to another in a manner opposite that for displacement. For example, if the velocity alarms for 2X and 3X RPM have been set at one half that at 1X RPM, he will need to set the acceleration alarm at 2X RPM at the same level as that at 1X RPM, but set the 3X alarm at 50% higher than that at 1X RPM. This is due to the fact that acceleration varies directly with both velocity and frequency (see formula in Figure 1).

7.13 REVIEW OF PROBLEMS DETECTABLE BY VIBRATION ANALYSIS

An essential need in specifying effective spectral alarm bands is a firm understanding of what problems are detectable by vibration analysis, how they are detected, and, if detected, how severe they are. Much research has been performed and much continues today on how to evaluate such things as balance condition, alignment, bearing health, gear health, electrical condition, etc. **"Illustrated Vibration Diagnostic Chart"** in Table I represents the best understanding to date of the author on how these problems are best diagnosed, based on much field experience and research of a wide range of articles which have been written on the subject. The list of references researched for this Diagnostic Chart should give an idea of the study which was required to prepare these tables (Reference nos. 2, 3, 4, 5, 6, 9, 11, 12, 15, 16, 17, 20, 21, 22, 23, 25, 26, 27, 29, 30, 31, 34, 35, 36, and 38).

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There are several key items included in Table I. First, the plots under "TYPICAL SPECTRUM" column reveal invaluable information as to the source of the problem:

1. Which frequencies are present in spectrum and how do they relate to machine speed (1X RPM)?

2. What are the amplitudes of each peak?

3. How do the frequency peaks relate to one another? (i.e., "2X RPM is much higher than 1X RPM"; "there is a large peak at 7.43X RPM"; "there are large number of operating speed harmonics present"; "there are high amplitude sidebands around gear mesh frequency"; etc.)

As its column name implies, "TYPICAL SPECTRUM" is meant to be a representative signature for each problem listed in Table I. These spectra are not intended to be all inclusive. For example, referring to "REMARKS" for the Angular Misalignment problem, note that while the typical spectrum shows high amplitude 1X RPM and 2X RPM peaks in the axial direction, the discussion shows that it is not unusual for either 1X, 2X or 3X RPM to dominate the whole spectrum. Similarly, this can occur with either radial misalignment or a cocked bearing. In addition, it is not unusual for a machine to have two or more problems at any one time. For example, if a machine simultaneously had both mechanical looseness and rotor unbalance, they each would be indicated in its spectra which would likely show high 1X RPM, in addition to multiple running speed harmonics.

The next column in Table I is entitled "PHASE RELATIONSHIP". Information on phase is provided for several of the problem sources listed. Amplitude reveals how much something is vibrating. Frequency relates how many cycles occur per unit of time. Phase completes the picture by showing just how the machine is vibrating. Of great importance, phase is a powerful tool in helping differentiate which of several problem sources are dominant. For example, there are a large number of problems that generate vibration at 1X and 2X RPM. Using phase, one learns how the machine is vibrating, and in the process, helps zero in on just which problem is present. For example, Table I shows:

1. Force (or "static") unbalance is evidenced by nearly identical phase in the radial direction on each bearing of a machine (outboard and inboard horizontals are in phase; outboard and inboard verticals are in phase).

2. Couple unbalance shows approximately a 180° out-of-phase relationship when comparing the outboard and inboard horizontal, or outboard and inboard vertical direction phase on the same machine rotor.

3. Angular misalignment is indicated by approximately a 180° phase difference across the coupling, with measurements in the axial direction.

4. Parallel misalignment causes radial direction phase across the coupling to be approximately 180° out of phase with respect to one another.

5. Bent shaft causes axial phase on the same shaft of a machine to approach 180° difference when comparing measurements on the outboard and inboard bearing of the same machine.

6. Resonance is shown by exactly a 90° phase change at the point when the forcing frequency coincides with a natural frequency, and approaches a full 180° phase change when the machine passes through the natural frequency (depending on the amount of damping present).

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7. Rotor rub causes significant, instantaneous changes in phase.

8. Mechanical looseness usually causes phase to be unsteady, with widely differing measurements from one time to the next. The phase measurement may noticeably differ every time you start up the machine, particularly if the rotor itself is loose and rotates on the shaft a few degrees with each startup.

Often, even though phase measurement capability is now offered by most data collectors, users do not use this powerful tool. If not used, this will severely limit the diagnostic capabilities of any program.

Note that “PHASE RELATIONSHIP” is illustrated in each of the first 8 problems of Table I since it is primarily with these problems that phase can be used to differentiate which problem(s) dominate. Phase is then discussed in many of the remarks for the remaining problems in Table I, although it is not illustrated.

Finally, a “REMARKS” column is included in Table I to provide further explanatory information on machine problem symptoms and diagnostics. For example, there is a warning under the remarks column for the “bent shaft” problem source to be sure and account for transducer orientation when taking axial phase measurements.

It is hoped that this diagnostic chart will help users in diagnosing a wide variety of machine problems. Further information is now being researched and field tested which may soon be added to the diagnostic chart.

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TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

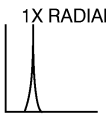
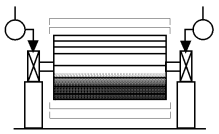
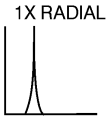
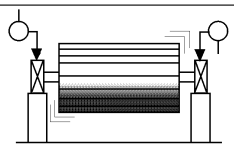
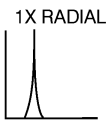
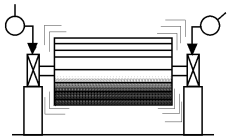
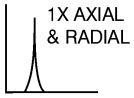
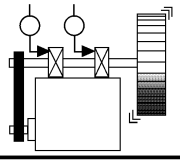
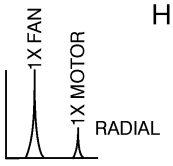
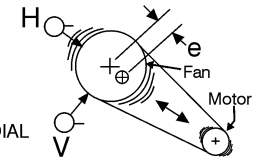
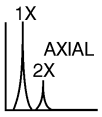
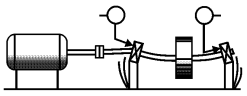
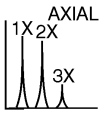
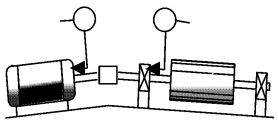
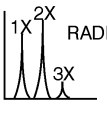
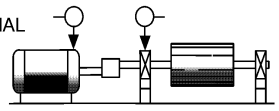
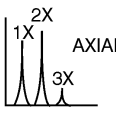
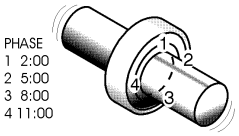
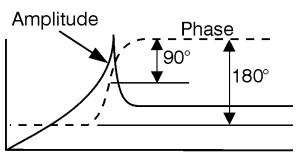
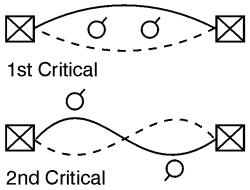
PROBLEM SOURCE	TYPICAL SPECTRUM	PHASE RELATIONSHIP	REMARKS
MASS UNBALANCE **A. FORCE UNBALANCE**			Force Unbalance will be in-phase and steady. Amplitude due to unbalance will increase by the square of speed below first rotor critical (a 3X speed increase = 9X higher vibration). 1X RPM always present and normally dominates spectrum. Can be corrected by placement of only one balance correction weight in one plane at Rotor center of gravity (CG). Approx. 0° phase difference should exist between OB & IB horizontals, as well as between OB & IB verticals. Also, approx. 90° phase difference between horizontal & vertical readings usually occurs on each bearing of unbalanced rotor (±30°).
B. COUPLE UNBALANCE			Couple Unbalance results in 180° out-of-phase motion on same shaft. 1X RPM always present and normally dominates spectrum. Amplitude varies with square of increasing speed below first rotor critical speed. May cause high axial vibration as well as radial. Correction requires placement of balance weights in at least 2 planes. Note that approx. 180° phase difference should exist between OB & IB horizontals, as well as between OB & IB verticals. Also, approx. a 90° difference between the horizontal & vertical phase readings on each bearing usually occurs (±30°)
C. DYNAMIC UNBALANCE			Dynamic Unbalance is the dominant type of unbalance found and is a combination of both force and couple unbalance. 1X RPM dominates the spectrum, and truly requires 2 plane correction. Here, the radial phase difference between outboard and inboard bearings can range anywhere from 0° to 180°. However, the horizontal phase difference should closely match the vertical phase difference, when comparing outboard and inboard bearing measurements (±30°). Secondarily, if unbalance predominates, roughly a 90° phase difference usually results between the horizontal and vertical readings on each bearing (±40°).
D. OVERHUNG ROTOR UNBALANCE			Overhung Rotor Unbalance causes high 1X RPM in both Axial and Radial directions. Axial readings tend to be in-phase whereas radial phase readings might be unsteady. However, the horizontal phase differences will usually match the vertical phase differences on the unbalanced rotor (±30°). Overhung rotors have both force and couple unbalance, each of which will likely require correction. Thus, correction weights will most always have to be placed in 2 planes to counteract both force and couple unbalance.
ECCENTRIC ROTOR			Eccentricity occurs when center of rotation is offset from geometric centerline of a pulley, gear, bearing, motor armature, etc. Largest vibration occurs at 1X RPM of eccentric component in a direction thru centerlines of the two rotors. Comparative horizontal and vertical phase readings usually differ either by 0° or by 180° (each of which indicate straight-line motion). Attempts to balance eccentric rotors often result in reducing vibration in one radial direction, but increasing it in the other radial direction (depending on amount of eccentricity).
BENT SHAFT			Bent shaft problems cause high axial vibration with axial phase differences tending towards 180° on the same machine component. Dominant vibration normally occurs at 1X if bent near shaft center, but at 2X if bent near the coupling. (Be careful to account for transducer orientation for each axial measurement if you reverse probe direction.) Use dial indicators to confirm bent shaft.
MISALIGNMENT **A. ANGULAR MISALIGNMENT**			Angular Misalignment is characterized by high axial vibration, 180° out-of-phase across the coupling. Typically will have high axial vibration with both 1X and 2X RPM. However, not unusual for either 1X, 2X or 3X to dominate. These symptoms may also indicate coupling problems as well. Severe angular misalignment may excite many 1X RPM harmonics. Unlike Mechanical Looseness Type 3, these multiple harmonics do not typically have a raised noise floor on the spectra.
B. PARALLEL MISALIGNMENT			Offset Misalignment has similar vibration symptoms to Angular, but shows high radial vibration which approaches 180° out-of-phase across coupling. 2X often larger than 1X, but its height relative to 1X is often dictated by coupling type and construction. When either Angular or Radial Misalignment becomes severe, they can generate either high amplitude peaks at much higher harmonics (4X-8X), or even a whole series of high frequency harmonics similar in appearance to mechanical looseness. Coupling type and material will often greatly influence the entire spectrum when misalignment is severe. Does not typically have raised noise floor.
C. MISALIGNED BEARING COCKED ON SHAFT			Cocked Bearing will generate considerable axial vibration. Will cause Twisting Motion with approximately 180° phase shift top to bottom and/or side to side as measured in axial direction on same bearing housing. Attempts to align coupling or balance the rotor will not alleviate problem. Bearing usually must be removed and correctly installed.
RESONANCE			Resonance occurs when a Forcing Frequency coincides with a System Natural Frequency, and can cause dramatic amplitude amplification, which might result in premature, or even catastrophic failure. This may be a natural frequency of the rotor, but can often originate from support frame, foundation, gearbox or even drive belts. If a rotor is at or near resonance, it can be almost impossible to balance due to the great phase shift it experiences (90° at resonance; nearly 180° when passes thru). Often requires changing natural frequency to a higher or lower frequency. Natural Frequencies do not generally change with a change in speed which helps facilitate their identification (unless on a large plain bearing machine or on a rotor which has significant overhang).

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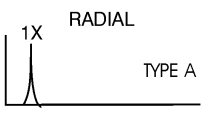
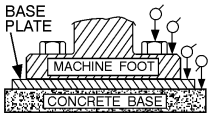
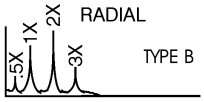
TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

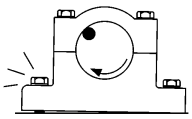
PROBLEM SOURCE	TYPICAL SPECTRUM	PHASE RELATIONSHIP	REMARKS

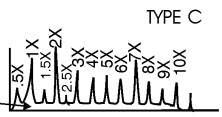
MECHANICAL LOOSENESS

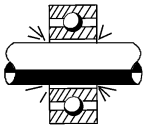






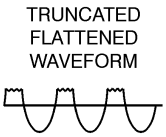


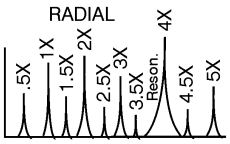




Mechanical Looseness is indicated by either Type A, B or C vibration spectra. ***Type A*** is caused by Structural looseness/weakness of machine feet, baseplate or foundation; also by deteriorated grouting, loose hold-down bolts at the base; and distortion of the frame or base (i.e., soft foot). Phase analysis may reveal approx. 90° to 180° phase difference between vertical measurements on bolt, machine foot, baseplate or base itself.
Type B is generally caused by loose pillowblock bolts, cracks in frame structure or in bearing pedestal.
Type C is normally generated by improper fit between component parts which will cause many harmonics due to nonlinear response of loose parts to dynamic forces from rotor. Causes a truncation of time waveform and a raised noise floor in the spectrum. Type C is often caused by a bearing liner loose in its cap, a bearing loose and turning on its shaft, excessive clearance in either a sleeve or rolling element bearing, a loose impeller on a shaft, etc. Type C Phase is often unstable and may vary widely from one measurement to next, particularly if rotor shifts position on shaft from one startup to next. Mechanical Looseness is often highly directional and may cause very different readings when comparing levels at 30° increments in radial direction all the way around one bearing housing. Also, note that looseness will often cause subharmonic multiples at exactly 1/2 or 1/3X RPM (.5X, 1.5X, 2.5X, etc.).

ROTOR RUB

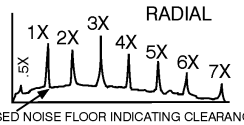




Rotor Rub produces similar spectra to Mechanical Looseness when rotating parts contact stationary components. Rub may be either partial or throughout the entire shaft revolution. Usually generates a series of frequencies, often exciting one or more resonances. Often excites integer fraction subharmonics of running speed (1/2, 1/3, 1/4, 1/5,...1/n), depending on location of rotor natural frequencies. Rotor rub can excite many high frequencies (similar to wide-band noise when chalk is drug along a blackboard). It can be very serious and of short duration if caused by shaft contacting bearing babbitt. A full annular rub throughout an entire shaft revolution can induce "reverse precession" with the rotor whirling at critical speed in a direction opposite shaft rotation (inherently unstable which can lead to catastrophic failure).

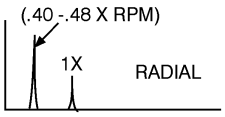
JOURNAL BEARINGS

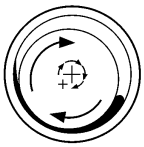
A. WEAR/CLEARANCE PROBLEMS



Latter stages of journal bearing wear are normally evidenced by presence of whole series of running speed harmonics (up to 10 or 20). Wiped journal bearings often will allow high vertical amplitudes compared to horizontal, but, may show only one pronounced peak at 1X RPM. Journal bearings with excessive clearance may allow a minor unbalance and/or misalignment to cause high vibration which would be much lower if bearing clearances were set to spec.

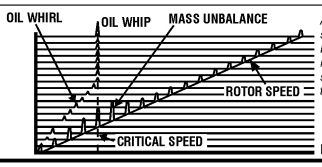
B. OIL WHIRL INSTABILITY





Oil Whirl instability occurs at .40 - .48X RPM and is often quite severe. Considered excessive when amplitude exceeds 40% of bearing clearances. Oil Whirl is an oil film excited vibration where deviations in normal operating conditions (attitude angle and eccentricity ratio) cause oil wedge to "push" shaft around within bearing. Destabilizing force in direction of rotation results in a whirl (or forwards precession). Oil Whirl is unstable since it increases centrifugal forces which increase whirl forces. Can cause oil to no longer support shaft and can become unstable when whirl frequency coincides with a rotor natural frequency. Changes in oil viscosity, lube pressure and external preloads can affect oil whirl.

C. OIL WHIP INSTABILITY

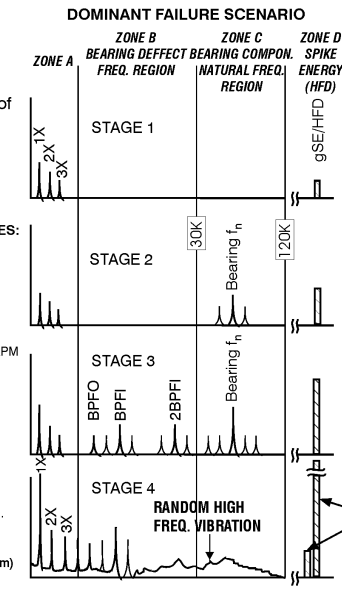


A spectral map showing Oil Whirl becoming Oil Whip instability as shaft speed passes thru 2X critical.

Oil Whip may occur if machine operated at or above 2X rotor critical frequency. When rotor brought up to twice critical speed, whirl will be very close to rotor critical and may cause excessive vibration that oil film may no longer be capable of supporting. Whirl speed will actually "lock onto" rotor critical and this peak will not pass through it even if machine is brought to higher and higher speeds. Produces a lateral forward precessional subharmonic vibration at rotor critical frequency. Inherently unstable which can lead to catastrophic failure.

ROLLING ELEMENT BEARINGS

(4 Failure Stages)



f_n = Natural Frequencies of Installed Bearing Components and Support Structure

BEARING DEFECT FREQUENCIES:

$$BPFI = \frac{N_b}{2}\left(1+\frac{B_d}{P_d}\cos\theta\right) \times RPM$$

$$BPFO = \frac{N_b}{2}\left(1-\frac{B_d}{P_d}\cos\theta\right) \times RPM$$

$$BSF = \frac{P_d}{2B_d}\left[1-\left(\frac{B_d}{P_d}\right)^2(\cos\theta)^2\right] \times RPM$$

$$FTF = \frac{1}{2}\left(1-\frac{B_d}{P_d}\cos\theta\right) \times RPM$$

Where:
BPFI = Inner Race Frequency
BPFO = Outer Race Frequency
BSF = Ball Spin Frequency
FTF = Fund. Train (Cage) Freq.
N_b = Number of Balls or Rollers
B_d = Ball/Roller Diameter (in or mm)
P_d = Bearing Pitch Diameter (in or mm)
θ = Contact Angle (degrees)

4 ROLLING ELEMENT BEARING FAILURE STAGES

STAGE 1: Earliest indications of bearing problems appear in ultrasonic frequencies ranging from about 250,000 - 350,000 Hz; later, as wear increases, usually drops to approximately 20,000 - 60,000 Hz (1,200,000 - 3,600,000 CPM). These are frequencies evaluated by Spike Energy (gSE), HFD(g) and Shock Pulse (dB). For example, spike energy may first appear at about .25 gSE in Stage 1 (actual value depending on measurement location and machine speed). Acquiring high frequency enveloped spectra confirms whether or not bearing is in Failure Stage 1.

STAGE 2: Slight bearing defects begin to "ring" bearing component natural frequencies (f_n) which predominantly occur in 30K - 120K CPM range. Such natural frequencies may also be resonances of bearing support structures. Sideband frequencies appear above and below natural frequency peak at end of Stage 2. Overall spike energy grows (for example, from .25 to .50 gSE).

STAGE 3: Bearing defect frequencies and harmonics appear. When wear progresses, more defect frequency harmonics appear and number of sidebands grow, both around these and bearing component natural frequencies. Overall spike energy continues to increase (for example, from .5 to over 1 gSE). Wear is now usually visible and may extend throughout periphery of bearing, particularly when many well formed sidebands accompany bearing defect frequency harmonics. High frequency demodulated and enveloped spectra help confirm Stage III. ***Replace bearings now! (independent of bearing defect frequency amplitudes in vibration spectra).***

STAGE 4: Towards the end, amplitude of 1X RPM is even effected. It grows, and normally causes growth of many running speed harmonics. Discrete bearing defect and component natural frequencies actually begin to "disappear" and are replaced by random, broadband high frequency "noise floor". In addition, amplitudes of both high frequency noise floor and spike energy may in fact decrease; but just prior to failure, spike energy and HFD will usually grow to excessive amplitudes.

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TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

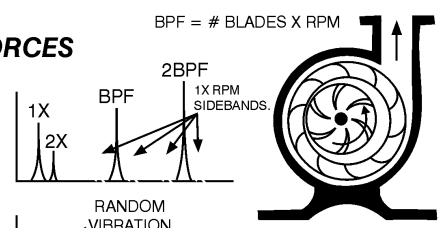
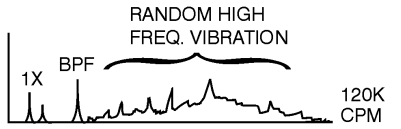
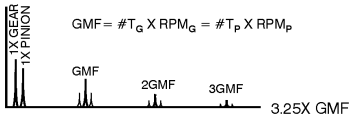
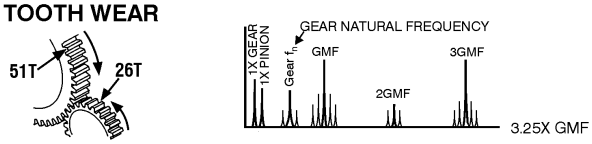
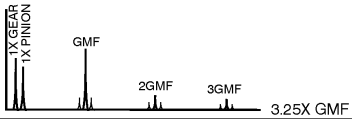
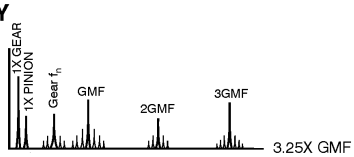
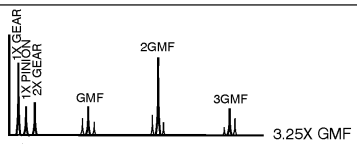
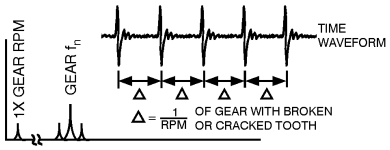
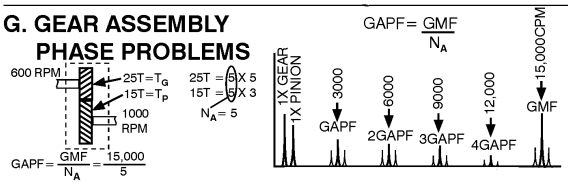
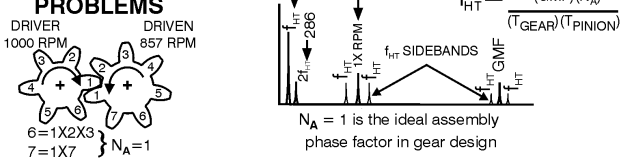
PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS
HYDRAULIC AND AERODYNAMIC FORCES		
A. BLADE PASS & VANE PASS		Blade Pass Frequency (BPF) = No. of Blades (or Vanes) X RPM. This frequency is inherent in pumps, fans and compressors, and normally does not present a problem. However, large amplitude BPF (and harmonics) can be generated in pump if gap between rotating vanes and stationary diffusers is not equal all the way around. Also, BPF (or harmonic) sometimes can coincide with a system natural frequency causing high vibration. High BPF can be generated if impeller wear ring seizes on shaft, or if welds fastening diffuser vanes fail. Also, high BPF can be caused by abrupt bends in pipe (or duct), obstructions which disturb flow, damper settings or if pump or fan rotor is positioned eccentrically within housing.
B. FLOW TURBULENCE		Flow Turbulence often occurs in blowers due to variations in pressure or velocity of the air passing thru the fan or connected ductwork. This flow disruption causes turbulence which will generate random, low frequency vibration, typically in the range of 50 to 2000 CPM. If surging occurs within a compressor, random broadband high frequency vibration can occur. Excessive turbulence can also excite broadband high frequency.
C. CAVITATION		Cavitation normally generates random, higher frequency broadband energy which is sometimes superimposed with blade pass frequency harmonics. Normally indicates insufficient suction pressure (starvation). Cavitation can be quite destructive to pump internals if left uncorrected. It can particularly erode impeller vanes. When present, it often sounds as if "gravel" is passing thru pump. Cavitation is usually caused by insufficient inlet flow. Can occur during one survey, and be absent the next survey (if changes in suction valve settings are made).
GEARS		
A. NORMAL SPECTRUM		Normal Spectrum shows Gear & Pinion Speeds, along with Gear Mesh Frequency (GMF) and very small GMF harmonics. GMF harmonics commonly will have running speed sidebands around them. All peaks are of low amplitude, and no natural frequencies of gears are excited. F_{MAX} recomended at 3.25X GMF (minimum) when # teeth are known. If tooth count is not known, set F_{MAX} at 200X RPM on each shaft.
B. TOOTH WEAR		Key indicator of Tooth Wear is excitation of Gear Natural Frequency (f_n), along with sidebands around it spaced at the running speed of the bad gear. Gear Mesh Frequency (GMF) may or may not change in amplitude, although high amplitude sidebands and number of sidebands surrounding GMF usually occur when wear is noticeable. Sidebands may be better wear indicator than GMF frequencies themselves. Also, high amplitudes commonly occur at either 2XGMF or at 3XGMF (esp. 3XGMF), even when GMF amplitude is acceptable.
C. TOOTH LOAD		Gear Mesh Frequencies are often very sensitive to load. High GMF amplitudes do not necessarily indicate a problem, particularly if sideband frequencies remain low level, and no gear natural frequencies are excited. Each Analysis should be performed with system at maximum operating load for meaningful spectral comparisons.
D. GEAR ECCENTRICITY AND BACKLASH		Fairly high amplitude sidebands around GMF harmonics often suggest gear eccentricity, backlash, or non-parallel shafts which allow the rotation of one gear to "modulate" either the GMF amplitude or the running speed of the other gear. The gear with the problem is indicated by the spacing of the sideband frequencies.Also, 1X RPM level of eccentric gear will normally be high if eccentricity is the dominant problem. Improper backlash normally excites GMF harmonics and Gear Natural Frequency, both of which will be sidebanded at 1X RPM. GMF amplitudes will often decrease with increasing load if backlash is the problem.
E. GEAR MISALIGNMENT		Gear Misalignment almost always excites second order or higher GMF harmonics which are sidebanded at running speed. Often will show only small amplitude 1X GMF, but much higher levels at 2X or 3X GMF. Important to set F_{MAX} high enough to capture at least 3 GMF harmonics. Also, sidebands around 2XGMF will often be spaced at 2X RPM. Note that sideband amplitudes often are not equal on left and right side of GMF and GMF harmonics due to the tooth misalignment. Causes uneven wear pattern.
F. CRACKED/BROKEN TOOTH		A Cracked or Broken Tooth will generate a high amplitude at 1X RPM of this gear only in the time waveform, plus it will excite gear natural frequency (f_n) sidebanded at its running speed. It is best detected in Time Waveform which will show a pronounced spike every time the problem tooth tries to mesh with teeth on the mating gear. Time between impacts (Δ) will correspond to 1/RPM of gear with the problem. Amplitudes of Impact Spikes in Time Waveform often will be 10X to 20X higher than that at 1X RPM in the FFT!
G. GEAR ASSEMBLY PHASE PROBLEMS	 GAPF= 3000 CPM = 0.20X GMF **(FRACTIONAL GMF)**	Gear Assembly Phase Freq. (GAPF) can result in Fractional Gear Mesh Frequencies (if N_A>1). It literally means (T_G/N_A) gear teeth will contact (T_P/N_A) pinion teeth and will generate N_A wear patterns, where N_A in a given tooth combination equals the product of prime factors common to the number of teeth on the gear and pinion (N_A = Assembly Phase Factor). GAPF (or harmonics) can show up right from the beginning if there were manufacturing problems. Also, its sudden appearance in a periodic survey spectrum can indicate damage if contaminate particles pass through the mesh, resulting in damage to the teeth in mesh at the time of ingestion just as they enter and leave meshing or that gears have been reoriented.
H. HUNTING TOOTH PROBLEMS	 f_{HT} = $\frac{(6X\ 1000)(1)}{(6)(7)}$ = $\frac{1000}{7}$ = 143 CPM (One Pulse Per 7 Pinion Revolutions)	Hunting Tooth Frequency (f_{HT}) occurs when faults are present on both the gear and pinion which might have occurred during the manufacturing process, due to mishandling, or in the field. It can cause quite high vibration, but since it occurs at low frequencies predominately less than 600 CPM, it is often missed. A gear set with this tooth repeat problem normally emits a "growling" sound from the drive. The maximum effect occurs when the faulty pinion and gear teeth both enter mesh at the same time (on some drives, this may occur only 1 of every 10 to 20 revolutions, depending on the f_{HT} formula). Note that T_{GEAR} and T_{PINION} refer to number of teeth on the gear and pinion, respectively. N_A is the Assembly Phase Factor defined above. Will often modulate both GMF and Gear RPM peaks.

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TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

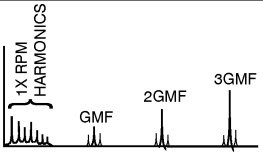
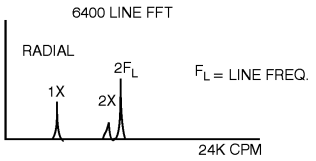
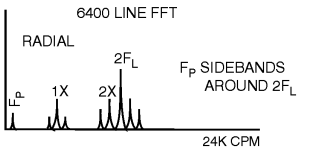
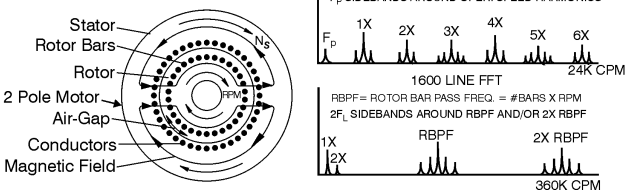
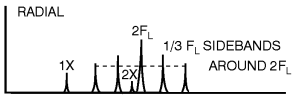
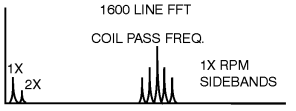
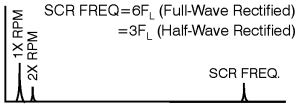
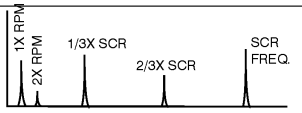
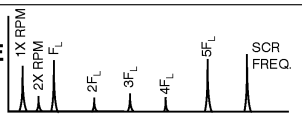
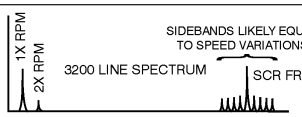
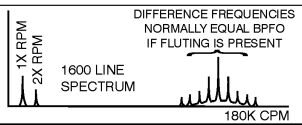
PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS
GEARS (CONTINUED)		
I. LOOSE BEARING FIT		Excessive Clearance of bearings supporting the gears can not only excite many running speed harmonics, but will often cause high amplitude response at GMF, 2GMF and/or 3GMF. These high GMF amplitudes are actually a response to, and not the cause of, looseness within the bearings supporting the gearing. Such excessive clearance can be caused either by extensive bearing wear or by improper bearing fit onto the journal during installation. Left uncorrected, it can cause excessive gear wear and damage to other components.
AC INDUCTION MOTORS		
A. STATOR ECCENTRICITY, SHORTED LAMINATIONS OR LOOSE IRON		Stator problems generate high vibration at 2X line frequency ($2F_L$). Stator eccentricity produces uneven stationary air gap between rotor and stator which produces very directional vibration. Differential Air Gap should not exceed 5% for induction motors and 10% for synchronous motors. Soft foot and warped bases can produce an eccentric stator. Loose iron is due to stator support weakness or looseness. Shorted stator laminations can cause uneven, localized heating which can distort the stator itself. This produces thermally-induced vibration which can significantly grow with operating time causing stator distortion and static air gap problems.
B. ECCENTRIC ROTOR (Variable Air Gap) F_L = Electrical Line Freq. N_S = Synch. Speed = $\frac{120F_L}{P}$ F_S = Slip Freq. = N_S – RPM F_P = Pole Pass Freq. = F_S X P P = #Poles		Eccentric Rotors produce a rotating variable air gap between the rotor and stator which induces pulsating vibration (normally between $2F_L$ and closest running speed harmonic). Often requires "zoom" spectrum to separate $2F_L$ and running speed harmonic. Eccentric rotors generate $2F_L$ surrounded by Pole Pass frequency sidebands (F_P), as well as F_P sidebands around running speed. F_P appears itself at low frequency (Pole Pass Frequency = Slip Frequency X #Poles). Common values of F_P range from about 20 to 120 CPM (0.3 - 2.0 Hz). Soft foot or misalignment often induces a variable air gap due to distortion (actually a mechanical problem; not electrical).
C. ROTOR PROBLEMS		Broken or Cracked rotor bars or shorting rings; bad joints between rotor bars and shorting rings; or shorted rotor laminations will produce high 1X running speed vibration with pole pass frequency sidebands (F_P). In addition, these problems will often generate F_P sidebands around the second, third, fourth and fifth running speed harmonics. Loose or open rotor bars are indicated by 2X line freq. ($2F_L$) sidebands surrounding Rotor Bar Pass Frequency (RBPF) and/or its harmonics (RBPF = Number of Bars X RPM). Often will cause high levels at 2X RBPF, with only a small amplitude at 1X RBPF. Electrically induced arcing between loose rotor bars and end rings will often show high levels at 2X RBPF (with $2F_L$ sidebands); but little or no increase in amplitudes at 1X RBPF.
D. PHASING PROBLEM (Loose Connector)		Phasing problems due to loose or broken connectors can cause excessive vibration at 2X Line Freq. ($2F_L$) which will have sidebands around it spaced at 1/3 Line Freq. (1/3 F_L). Levels at $2F_L$ can exceed 1.0 in/sec if left uncorrected. This is particularly a problem if the defective connector is only sporadically making contact. Loose or broken connectors must be repaired to prevent catastrophic failure.
AC SYNCHRONOUS MOTORS **(Loose Stator Coils)**		Loose stator coils in synchronous motors will generate fairly high vibration at Coil Pass Freq. (CPF) which equals the number of stator coils X RPM (#Stator Coils = #Poles X #Coils/Pole). The Coil Pass Frequency will be surrounded by 1X RPM sidebands. Synchronous motor problems may also be indicated by high amplitude peaks at approx. 60,000 to 90,000 CPM, accompanied by $2F_L$ sidebands. Take at least one spectrum up to 90,000 CPM on each motor bearing housing.
DC MOTORS AND CONTROLS **A. NORMAL SPECTRUM**		Many DC Motor and Control Problems can be detected by vibration analysis. Full-wave rectified, motors (6 SCR's) generate a signal at 6X Line Frequency ($6F_L$=360 Hz=21,600 CPM); while half-wave rectified DC motors (3 SCR's) generate 3X Line Freq. ($3F_L$=180 Hz=10,800 CPM). The SCR firing Frequency is normally present in a DC Motor Spectrum, but at low amplitude. Note the absence of other peaks at multiples of F_L.
B. BROKEN ARMATURE WINDINGS, GROUNDING PROBLEMS OR FAULTY SYSTEM TUNING		When DC Motor spectra are dominated by high levels at SCR or 2X SCR, this normally indicates either Broken Motor Windings or Faulty Tuning of the Electrical Control System. Proper tuning alone can lower vibration at SCR and 2X SCR significantly if control problems predominate. High amplitudes at these frequencies would normally be above approximately .10 in/sec, peak at 1 X SCR and about .04 in/sec at 2 X SCR Firing Freq.
C. FAULTY FIRING CARD OR BLOWN FUSE		When one firing card fails to fire, then 1/3 of power is lost, and can cause repeated momentary speed changes in the motor. This can lead to high amplitudes at 1/3X and 2/3X SCR Frequency (1/3X SCR Freq. = 1X F_L for half-wave rectified, but 2X F_L for a full-wave rectified SCR). Caution: Card/SCR configuration should be known before troubleshooting motor (#SCR's, #Firing Cards, etc.).
D. FAULTY SCR, SHORTED CONTROL CARD, LOOSE CONNECTIONS AND/OR BLOWN FUSE		Faulty SCR's, Shorted Control Cards and/or Loose Connections can generate noticeable amplitude peaks at many combinations of line frequency (F_L) and SCR firing frequency. Normally, 1 bad SCR can cause high levels at F_L and/or $5F_L$ in 6 SCR motors. The point to be made is that neither F_L, $2F_L$, $4F_L$ nor $5F_L$ should be present in DC Motor spectra.
E. FAULTY COMPARITOR CARD		Faulty Comparitor Cards cause problems with RPM fluctuation or "hunting". This causes a constant collapsing and regenerating of the magnetic field. These sidebands often approximate the RPM fluctuation and require a high resolution FFT to even detect them. Such sidebands could also be due to generation and regeneration of the magnetic field.
F. ELECTRICAL CURRENT PASSAGE THRU DC MOTOR BEARINGS		Electrically-induced Fluting is normally detected by a series of difference frequencies with the spacing most often at the outer race defect frequency (BPFO), even if such fluting is present on both the outer and inner races. They most often show up in a range centered at about 100,000 to 150,000 CPM. A 180K CPM spectrum with 1600 lines is recommended for detection with measurements on both the OB and IB DC motor bearings.

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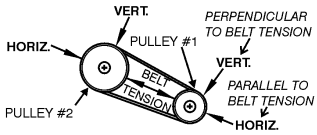
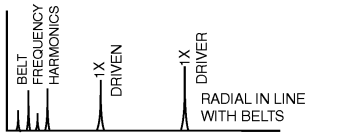
TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS

BELT DRIVE PROBLEMS

A. WORN, LOOSE OR MISMATCHED BELTS





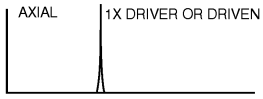
PITCH DIAM$_1$ X RPM$_1$ = PITCH DIAM$_2$ X RPM$_2$

BELT FREQ. = $\frac{3.142 \text{ X PULLEY RPM X PITCH DIAM.}}{\text{BELT LENGTH}}$

TIMING BELT FREQ. = BELT FREQ. X #BELT TEETH
= PULLEY RPM X #PULLEY TEETH

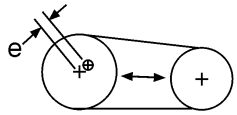
Belt frequencies are below the RPM of either the motor or the driven machine. When they are worn, loose or mismatched, they normally cause 3 to 4 multiples of belt frequency. Often 2X belt freq. is the dominant peak. Amplitudes are normally unsteady, sometimes pulsing with either driver or driven RPM. On timing belt drives, wear or pulley misalignment is indicated by high amplitudes at the Timing Belt Frequency. Chain drives will indicate problems at Chain Pass Frequency which equals #Sprocket Teeth X RPM.

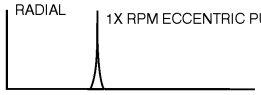
B. BELT/PULLEY MISALIGNMENT



Misalignment of pulley produces high vibration at 1X RPM predominantly in the axial direction. The ratio of amplitudes of driver to driven RPM depends on where the data is taken, as well as on relative mass and frame stiffness. Often with pulley misalignment, the highest axial vibration on the motor will be at fan RPM, or vice versa. Can be confirmed by phase measurements by setting Phase Filter at RPM of pulley with highest axial amplitude; then compare phase at this particular frequency on each rotor in the axial direction.

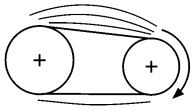
C. ECCENTRIC PULLEYS

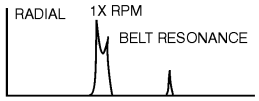




Eccentric pulleys cause high vibration at 1X RPM of the eccentric pulley. The amplitude is normally highest in line with the belts, and should show up on both driver and driven bearings. It is sometimes possible to balance eccentric pulleys by attaching washers to taper-lock bolts. However, even if balanced, the eccentricity will still induce vibration and reversible fatigue stresses in the belt. Pulley eccentricity can be confirmed by phase analysis showing horizontal & vertical phase differences of nearly 0° or 180°.

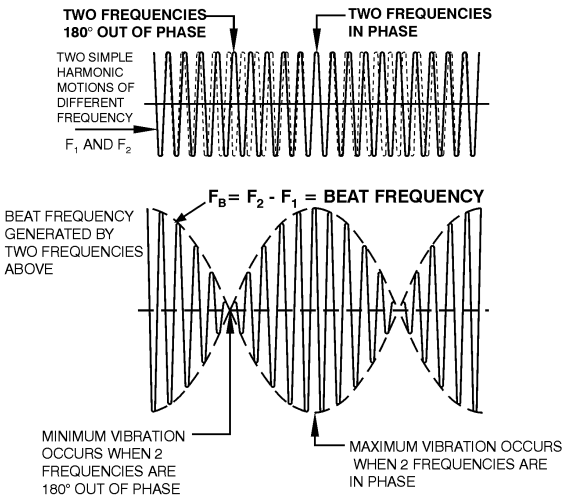
D. BELT RESONANCE





Belt Resonance can cause high amplitudes if the belt natural frequency should happen to approach, or coincide with, either the motor or driven RPM. Belt natural frequency can be altered by changing either the belt tension, belt length or cross section. Can be detected by tensioning and then releasing belt while measuring the response on pulleys or bearings. However, when operating, belt natural frequencies will tend to be slightly higher on the tight side and lower on the slack side.

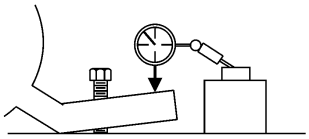
BEAT VIBRATION

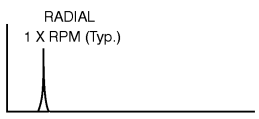

TWO FREQUENCIES 180° OUT OF PHASE
TWO FREQUENCIES IN PHASE
TWO SIMPLE HARMONIC MOTIONS OF DIFFERENT FREQUENCY
F1 AND F2
FB= F2 - F1 = BEAT FREQUENCY
BEAT FREQUENCY GENERATED BY TWO FREQUENCIES ABOVE
MINIMUM VIBRATION OCCURS WHEN 2 FREQUENCIES ARE 180° OUT OF PHASE
MAXIMUM VIBRATION OCCURS WHEN 2 FREQUENCIES ARE IN PHASE

A Beat Frequency is the result of two closely spaced frequencies going into and out of synchronization with one another. The wideband spectrum normally will show one peak pulsating up and down. When you zoom into this peak (lower spectrum below), it actually shows two closely spaced peaks. The difference in these two peaks (F_2-F_1) is the beat frequency which appears itself in the wideband spectrum. The beat frequency is not commonly seen in normal frequency range measurements since it is inherently low frequency, usually ranging from only approximately 5 to 100 CPM.

Maximum vibration will result when the time waveform of one frequency (F_1) comes into phase with the waveform of the other frequency (F_2). Minimum vibration occurs when waveforms of these two frequencies line up 180° out of phase.

SOFT FOOT, SPRUNG FOOT AND FOOT-RELATED RESONANCE





"***Soft Foot***" occurs when a machine's foot or frame deflects greatly when a hold-down bolt is loosened to hand tightness, causing the foot to rise more than approximately .002 - .003 inch. This does not always cause a great vibration increase. However, it can do so if the soft foot affects alignment or motor air gap concentricity.

"***Sprung Foot***" can cause great frame distortion, resulting in increased vibration, force and stress in the frame, bearing housing, etc. This can occur when a hold-down bolt is forceably torqued down on the sprung foot in an attempt to level the foot.

"***Foot-Related Resonance***" can cause dramatic amplitude increases from 5X to 15X or more, as compared with that when the bolt (or combination of bolts) is loosened to hand tightness. When tight, this bolt can notably change the natural frequency of the foot or machine frame itself.

Soft Foot, Sprung Foot or Foot-Related Resonance most often affects vibration at 1X RPM, but can also do so at 2X RPM, 3X RPM, 2X line frequency, blade pass frequency, etc. (particularly Foot-Related Resonance).

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7.14 SPECIFICATION OF OVERALL VIBRATION ALARM LEVELS AND EXPLANATION OF THE ORIGIN OF TABLE II "OVERALL CONDITION RATING" CHART

Much work continues today on establishing standards for allowable overall vibration. Various national and international committees made up of experienced professionals have been established and are given the charge of formulating these vibration criteria. This includes the international working group on machinery vibration standards which is now working to update several criteria (19):

ISO 2372 - "Mechanical Vibration of Machines with Operating Speeds from 10 to 200 Revolutions per Second" - Basis for specifying evaluation standards (measurements made on structure).

ISO 3945 - "Mechanical Vibration of Large Rotating Machines with Speeds Ranging from 10 to 200 Revolutions per Second" - Measurement and evaluation of vibration severity in situ (measurements made on structure at various elevations).

ISO 7919 - "Mechanical Vibration of Non-Reciprocating Machines" - Measurement on rotating components and evaluation (measurements made on shafting).

Some attempts have been made, and some are now being offered, to provide vibration criteria based on the type of machine and its drive configuration (centrifugal pump, direct coupled fan, belt driven fan, turbine/generator, etc.), and on its mounting (isolated versus non-isolated). It is recognized that there is often dramatic difference in the amount of vibration from one machine type versus another. For example, a reciprocating air compressor obviously has significantly more inherent vibration than does a hermetic chiller or, for that matter, a precision machine tool spindle. Also, a machine will generally experience higher vibration than before when placed on isolators, depending on the isolator type, isolator connection, isolator natural frequency, forcing frequencies of the machine itself, machine center of gravity relative to the placement of isolators, etc. Thus, it is important that the user of today's predictive maintenance hardware and software take into account both the type of machine and its mounting when he begins to specify alarm levels of overall vibration for each machine that he will input into his computer database.

In addition, it is important for the user to know how his particular predictive maintenance data collector and software system measures overall vibration. Some systems have a fixed frequency range, completely independent of any frequency range chosen on any particular spectrum. In fact, this overall measurement is completely independent from spectral measurements in some systems. In others, the overall is determined by first taking a spectrum, and then by using Equation (1) to calculate the spectral overall level [Equation (1) is repeated here for the reader]:

$$OA = \frac{\sqrt{\sum_{i=1}^{n} A_i^2}}{\sqrt{N_{BF}}} = \frac{\sqrt{A_1^2 + A_2^2 + A_3^2 + ... + A_N^2}}{\sqrt{1.5}}$$

Where:

OA = Overall Level of Spectrum or Band
n = Number of FFT Lines of Resolution
A_1 = Amplitude of Each of the FFT Lines
N_{BF} = Noise Bandwidth for Window Chosen
1.5 for Hanning Window

Which means $OA = .8165 \sqrt{\sum_{i=1}^{n} A_i^2}$

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During the years, our company has had the opportunity of analyzing a diverse array of both process and utility machinery ranging from very small, precision, high-speed spindles to large, slow moving machines. In addition, this work has been performed for a wide spectrum of industry types. This has given us invaluable exposure which has been greatly beneficial when opportunities have arisen for setting up predictive maintenance programs for these same clients.

Through the years, we have developed in-house vibration criteria specifically for the purpose of setting up these predictive maintenance databases. Some of the criteria we have developed is included in Table II. Note that Table II includes overall peak velocity criteria for measurements taken on the machine structure. Importantly, levels specified in Table II are not meant to be final, concrete numbers, but are intended to be a starting point when nothing is known about a machine other than its nameplate data, machine type and mounting. Later, after taking actual measurements on each point of each machine, these levels are individually reviewed and adjusted as needed. This refinement procedure is discussed later in the paper and examples are given illustrating the procedure.

Note that each of four "ratings" are provided in Table II including "GOOD", "FAIR", "ALARM 1" and "ALARM 2". After review of all spectra captured on a machine, if no problems are found, the first two columns ("GOOD" and "FAIR") are offered to give the client a general feel for the overall condition of each machine based on the highest overall level measured on his machine. However, even if the highest overall on a machine might remain within the "GOOD" range, it is still possible for the machine to be in alarm, depending on what frequencies were generated, the amplitudes of those frequencies, and the problem source(s) generating these frequencies. That is where the spectral alarm bands come into play to ferret out the "apparently good condition" machines from those that truly are in this condition. Corrective actions should be taken on those machines having vibration exceeding "ALARM 1"; while those exceeding "ALARM 2" are felt to be exposed to such high levels as to render potentially catastrophic failure (therefore, demanding immediate attention). Amplitudes listed in Table II were developed by calculating both the average level and standard deviation of large numbers of diverse types of machinery over a period of approximately 15 years in carrying out predictive maintenance programs. Then, "ALARM 1" overall levels were calculated by summing the average level plus 3 times the standard deviation [see Equations (2) through (4B)]. Final statistical overall levels were then rounded to the nearest ".025" level (that is, a level of .318 would be rounded off to .325 in/sec). Finally "ALARM 2" levels were determined by increasing "ALARM 1" levels by 50%.

Importantly, not only do the overall levels specified in Table II serve as an overall alarm given in the PMP database, but also they are used as direct input for specifying alarm levels for each specific spectral band in the section which will follow. Of course, if this overall is later refined after making several measurements on a machine or on a group of machines, the spectral band alarm levels themselves will also have to be adjusted as well.

7.15 SPECIFICATION OF SPECTRAL ALARM LEVELS AND FREQUENCY BANDS USING TABLE III

Table III provides the procedure on how to originally specify spectral alarm bands for various machine types and configurations using those types of predictive maintenance software systems which allow the spectrum to be broken up into 6 individual bands. Each of these bands can be set at any span of frequencies, and at any alarm level for each individual band as chosen by the user. Therein lies both the strength of spectral alarm bands, and paradoxically, their major weakness if the user himself does not know where each frequency span should be specified, nor how high to set each one of the band alarm levels. Therefore, the express purpose of this section is to provide the predictive maintenance software user with the capability, not only of originally setting up a PMP database using proper spectral alarm bands, but also to help him refine his database on which he might have been taking

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measurements for several years. Several years ago, our company made a detailed in-house study on how to specify these bands. At the conclusion of this study, we elected to develop a written, tabular procedure on how to properly specify them. Since that time, we have helped a number of clients set up bands on their specific machinery in their particular industry. In so doing, we have continued to learn more and more about how to best use them, and have "polished" and refined our techniques several times. In addition, much study went into preparation of Table III as can be seen by the list of references (see Reference nos. 3, 6, 8, 10, 11, 12, 13, 14, 18, 23, 26, 28, 29, 32, and 37).

Importantly, please note that the procedures specified in Table III assume casing measurements of peak velocity (in/sec) using instruments which measure RMS and "convert" them to peak levels by electronic multiplication of amplitudes by 1.414. This now includes most of the data collectors in use in the United States. Also, Table III specifies spectral bands whose alarm levels are compared to the total power within the band (so-called "Power Bands"). Please refer to the section entitled "Two Types of Spectral Alarm Bands"). Although Table III applies to peak velocity amplitudes, the reader can modify it for RMS simply by multiplying amplitudes by .707. Then, if he wishes to have them expressed in metric units (mm/sec), he can multiply these RMS in/sec amplitudes by a factor of 25.4 and rounding them to the nearest appropriate metric level.

Table III shows how spectral alarm bands are set up for a number of machine types and configurations. Cases A and B are for both the driver and driven components of general rotating machines which are outfitted with rolling element and sleeve bearings, respectively. Cases C and D specify high frequency measurement points which are to be taken on gearbox housings in close proximity to each gear mesh, and which are essential to evaluate the health and alignment of gearing. Case C assumes one knows the number of gear teeth, while Case D shows how to specify alarm bands for gearboxes where the number of teeth is unknown. Cases E and F are special points with the purpose of detecting potential motor electrical problems. The point specified by Case E is intended to detect the first and second harmonic rotor bar pass frequencies (number of bars X RPM), whereas the point specified by Case F attempts to separate mechanical from electrical vibration sources, particularly in the vicinity of machine operating speed, electrical synchronous frequency (60 Hz), and twice synchronous frequency (120 Hz). A number of special machine types are covered in Case G, which is intended to specify more specific alarm bands for these particular machine types.

Importantly, the specifications procedure outlined in Table III applies to general process and utility machinery such as centrifugal pumps, blowers, motors, forced-draft fans, induction-draft fans, motor/generator sets, centrifugal air compressors, refrigeration chillers, vacuum pumps, boiler feed pumps, gearboxes, etc. These specs do not apply to more specialized machine types such as reciprocating or rotary screw compressors; diesel engines; gas turbines; large turbine/generators; exciters; lobe-type rotary blowers; pulverizers; etc. Normally, spectral bands for these machine types have to be "custom-designed" for each set or grouping of them, and even then, often require the capture of several sets of data before one can begin to establish meaningful alarm bands.

For example, lobe-type rotary blowers (i.e., "Roots Blowers") present a real problem to the user who attempts to specify one all-encompassing set of alarm bands. They are offered in a wide range of sizes and configurations. Often, even after several surveys are conducted on these machines, the user may have difficulty in adequately specifying alarm bands since even identically sized and driven rotary blowers still can exhibit unique sets of vibration spectra (23). In reality, only 6 spectral alarm bands cannot adequately address these machines. They need approximately 10 to 12 bands to adequately cover them.

However, if the user is given the assignment of specifying spectral alarm bands for his plant, either when originally setting up its database or after several years of data have been captured (without adequate alarm bands), the procedure given in Table III should cover a large percentage of these

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machines. Before entering Table III, the user should identify his particular machine type and refer to Table II to find the alarm level of overall vibration for this machine. This will be used as direct input into the spectral alarm band specs of Table III. If his particular machine type is not included in Table II, the user should either refer to the manufacturer of his machine, other similar vibration severity charts, or use alarm levels for another machine type listed in Table II which most closely resembles his machine.

Please refer to the entries under the first column of Table III. "BAND LOWER FREQUENCY" specifies at what frequency each band should begin, whereas "BAND HIGHER FREQUENCY" shows where each band should end (for example, "from 60 to 1000 CPM"). In general, no gaps should be left between bands, nor should bands overlap one another (although some analysts using power bands sometimes extend one band from the beginning to the end of a complete spectrum in order to have the system calculate the "Spectral Overall Level", and then compare this to the overall level provided separately by their instrument). Next, the column entry entitled "BAND ALARM" specifies how high to set the alarm level of each band.

Notice that many of the cases described in Table III have the "BAND LOWER FREQUENCY" set at 1% F_{MAX} rather than at 0 CPM. The reason for this is that data collectors and spectrum analyzers most always have built-in "noise" within the first 1 to 3 FFT lines, particularly when data from an accelerometer are electronically integrated to velocity. In fact, some instruments have been known to display "peaks" with so-called "amplitudes" over 2.0 in/sec within these first 3 FFT lines. If F_{MAX} is properly specified, the first 2 to 3 FFT lines will almost always be contaminated with such electronic and/or integration "noise". Therefore, Band 1 will never begin within these first 3 lines in Table III.

Each of the cases specify the maximum frequency (F_{MAX}) which is always given along with the case title. Therefore, each case will tell where to set both the frequency range and alarm level of each band, and will describe what each band covers (i.e., bearing defect frequencies, gear mesh frequencies, etc.). Case A will be discussed in detail to illustrate the alarm band specification technique, whereas only highlights of each remaining case will be given. Then, several examples will follow the discussions to further illustrate how these techniques should be applied.

Case A - General Rolling Element Bearing Machine Without Rotating Vanes: (Motors, Spindles, Gearbox Lower Frequency Measurements, etc.)

Case A applies to both the driver and driven components of a wide range of general rotating process and utility machines which are outfitted with rolling element bearings (ball, roller or needle bearings). Before entering Table III, refer to Table II to obtain the alarm level of overall peak velocity for your machine type. Then, determine the type of rolling element bearing. For common rolling element bearings, Case A specifies a spectrum with a maximum frequency (F_{MAX}) of approximately 40X RPM (for example, for a nominal speed of 1800 RPM, set F_{MAX} at approximately 72,000 CPM). However, for tapered roller bearings (Timken cup and cone arrangement, or equivalent) or for spherical roller bearings, Case A specifies a maximum frequency of approximately 50X RPM. The reason for the higher F_{MAX} for these bearing types is the fact that, with their particular geometries, they inherently have higher calculated rolling element bearing defect frequencies. Also note that if the speed is below 1700 RPM, F_{MAX} must be set higher than 40X RPM (as seen in notes of Case A). The reason for this is to ensure that the spectra designed for this machine will detect a rolling element bearing in only the second of four failure stages through which it will normally pass rather than waiting late in the life of the bearing before problems are detected. Referring to Table I for "Rolling Element Bearings", note that the natural frequencies of bearing components will be excited during this second stage. Since these natural frequencies normally range from 30,000 to 120,000 CPM for most bearings, it is important to keep F_{MAX} sufficiently high to detect these when excited.
Please note that it is not necessary to specify F_{MAX} at exactly 40X or 50X RPM, but it should be

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somewhere in this vicinity (certainly not less than 35X RPM). If one sets F_{MAX} too low, it can cause a spectrum to completely miss potentially serious developing bearing wear, particularly during earlier stages. On the other hand, if F_{MAX} is set too high, this can result in poor frequency resolution which can cause the user to misdiagnose problems, since he does not have sufficiently precise frequency resolution to properly identify such frequency components as true running speed harmonics versus bearing defect frequencies, or vibration transmitting from adjacent machines. Also, if one sets F_{MAX} too high, this can cause potentially valuable information on subsynchronous vibration to be "buried" at the left-hand side of the spectrum. In general, the rule of thumb is to keep F_{MAX} as low as you can "without missing anything important".

Referring to Case A in Table III, note that each one of the bands has a specific purpose and zone of coverage. For example, Band 1 ranges from subsynchronous vibration (below 1X RPM) up through operating speed. Bands 2 and 3 cover 2X and 3X RPM, respectively. Band 4 will include fundamental bearing defect frequencies for most rolling element bearings. Similarly, Bands 5 and 6 will include bearing defect frequency harmonics, as well as natural frequencies of bearing components for most common rolling element bearings.

Now, referring back to Table III, note that Band 1 extends from 1% of F_{MAX} to a frequency at 1.2X RPM. In the case of the example 1800 RPM machine shown in Figure 2 having F_{MAX} at 72,000 CPM, Band 1 would extend from 720 to 2160 CPM. The Band 1 alarm spec calls for 90% of the overall level. Thus, if the overall alarm were .300 in/sec (from Table II), then the Band 1 alarm would be set at .270 in/sec for this machine. Similarly, Table III specifies the frequency range of Band 2 to extend from 1.2 to 2.2X RPM (in the 1800 RPM case, this would extend from approximately 2160 to 3960 CPM). The Band 2 alarm spec calls for 40% of the overall alarm (thus for the example .300 in/sec overall, Band 2 would be set at .120 in/sec). Finally, Bands 3 through 6 are specified similarly.

Case B - General Sleeve Bearing Machine Without Rotating Vanes:
(Sleeve Bearing Motors, Gearbox Lower Frequency Measurements, etc.)

Case B is similar to Case A, but is for general machines outfitted with sleeve bearings. Incidentally, if a sleeve bearing motor is driving a rolling element machine, Case A (rolling element) would be used for the driven machine points, whereas Case B (sleeve bearing) would be applied to the points on the motor.

Notice that F_{MAX} for these sleeve bearing machines is set only at 20X RPM as compared to 40X up to 100X RPM on rolling element bearing machines which inherently have much higher frequency spectra. In addition, some potentially serious problems can occur at subsynchronous frequencies on sleeve bearings including such things as oil whirl and oil whip. Therefore, this subsynchronous band needs to have good frequency resolution and must be closely watched.

Band 1 covers only the subsynchronous vibration in Case B while Bands 2, 3 and 4 include 1X, 2X and 3X RPM peaks, respectively (see Table III). Band 5 covers the range from 4X through 10X RPM while Band 6 extends from 10.5X RPM to F_{MAX}. Here again, the highest alarm level specified for any of the bands in Case B will be that at 1X RPM (Band 2). On the other hand, little amplitude is allowed in Band 6 even though it covers about 50% of the entire spectrum since only insignificant vibration should occur in this region if problems are not present, particularly if this machine is not a gearbox or connected to a gearbox.

Case C - Gearbox High Frequency Points with Known Number of Teeth:

Gearboxes require <u>two</u> sets of measurements on the same points due to the fact that many gear problems are detected at very high frequencies as compared to vibration due to such problems as unbalance, misalignment, etc. Therefore, one set of measurements should be taken on the

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gearbox using either Case A or B, depending on whether the gearbox is outfitted with rolling element (Case A) or sleeve (Case B) bearings. Then, a second set of measurements should be taken at various gearbox points close to each mesh, with F_{MAX} on this second measurement then set at 3.25X gear mesh frequency as shown in Table III. Very commonly, gearboxes may show low amplitudes at the fundamental gear mesh frequency (GMF), but may display very high amplitudes at 2X and/or 3X GMF. In addition, looseness is sometimes evidenced at one-half harmonics of gear mesh frequency, up to 2.5X GMF. Therefore, the maximum frequency is set at 3.25X GMF in order to allow for capture of gear mesh and accompanying sideband frequencies up thru 3X GMF.

Please note in Table III that spectra with 1600 to 3200 lines of resolution are recommended for these high frequency measurements. The reason for this is to allow 1X RPM sidebands to be displayed with good resolution around gear mesh frequency harmonics, not only for the high speed pinion, but also for the lower speed gear. Such high resolution spectra will also be recommended for Case D (when the tooth count is unknown).

A complete example illustrating specification of spectral alarm bands for a 2-stage speed increaser gearbox driving a compressor is given in Figure 3. Note the setups for both the lower frequency measurements (i.e., positions 3HI Axial and 3HI Horizontal) in Figure 3.

Please note the caution under Case C to keep in mind that a requirement to set F_{MAX} at 3.25X GMF may cause one to specify a maximum frequency that is not necessarily greater than the transducer frequency specifications, but can easily approach the natural frequency of the transducer mounting itself, causing errors in amplitude measurements. That is, when a transducer is mounted on a machine, it just creates another “spring/mass” in the system. The natural frequency of this spring/mass depends on how the transducer is mounted on the machine (stud, magnet, hand-held or extension probe). Stud mounting provides the highest natural frequency and, therefore, provides the highest measurable frequency with little or no deviation in amplitude readout. If forcing frequencies (such as gear mesh frequencies) are present close to the mounting natural frequency, considerable amplification can occur causing error in the amplitude readout, but not in the frequency. On the other hand, if forcing frequencies are above the mounting natural frequency, they can result in not only deviation in amplitude readout, but can also cause phase error since this system will experience almost a 180° difference in phase when it passes through resonance.

However, if this is kept in mind by the user, he can still take data at fairly high frequencies, being aware that amplitude levels may not be absolute. In any case, if they are repeatable, they can at least be trended; and, since frequency information remains correct, will likely allow the user to detect potential problems. If they are not repeatable, he will have to try a different transducer or method of mounting the original transducer.

Referring again to Case C in Table III, each of the frequency ranges are specified in terms of GMF multiples (for example, Band 2 extends from .75X GMF to 1.25X GMF). Here again, band alarms are set in terms of overall alarm percentage.

Importantly, if the gearbox has more than one set of individual meshes, as in the case of a double or a triple reduction unit, each set of high frequency points will need to employ the gear mesh frequency that applies at that particular measurement point. For example, if point A were close to the input gear mesh having a 100,000 CPM GMF and point B was at the output near a second mesh with a 25,000 CPM GMF, the high frequency point A would use the 100,000 CPM when setting up its bands (setting F_{MAX} at 325,000 CPM), whereas point B would employ the 25,000 CPM GMF (setting its F_{MAX} at approximately 81,250 CPM). This will be further illustrated in a gearbox example to be given later (Figure 3).

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Case D - Gearbox High Frequency Points with Unknown Number of Teeth:

Unfortunately, in most programs, the number of teeth in the great majority of gearboxes is unknown. In many cases, even the operating speeds of intermediate gears are unknown. However, in spite of this, one can set up effective spectral alarm bands which can be used until the true number of teeth are confirmed (however, when the tooth count is found, the spectral alarm bands should be respecified as per Case C). Referring to Case D in Table III, note that a maximum frequency of 200X shaft RPM will apply to each high frequency gearbox point. Note that the shaft speed at each particular measurement point will be used in specifying frequency ranges for each of the 6 bands. For example, if the input speed at point A was 1000 RPM and the output speed at point B was about 200 RPM, F_{MAX} at point A would be set at 1000 RPM X 200 (200,000 CPM) while that at point B would be set at 200 RPM X 200 (40,000 CPM).

In many plants, both the number of teeth and intermediate speeds are unknown in many multi-stage gearboxes. One approach to this problem of determining what the gear mesh frequencies are might be to acquire several sets of spectra on the gearbox and compare them to the "TYPICAL SPECTRUM" shown in Table I, Case E for "Gear Misalignment" (note that it shows both GMF and 2GMF, each of which are sidebanded at 1X RPM). If two or three harmonics of a high frequency fundamental are found (for example, fundamental at 40 to 60X RPM), it is possible that these are gear mesh frequencies, particularly if they each have 1X RPM sidebands. However, one must keep in mind that this same signature pattern could be caused by another problem (for example, rolling element bearing frequency harmonics at, say, 5X and 10X inner race frequency).

Therefore, Table III, Case D suggests another approach if the number of teeth and intermediate speeds are unknown. In these cases, one normally knows at least the gearbox ratio, and therefore, the input and output speeds. The note in Case D shows how to handle this case in which equal speed increment steps are assumed until one knows more about the intermediate shaft speeds. For example, if all you knew were the input speed, output speed and gear ratio, use the following formula as a start until you know more:

$$\text{Speed Increment Factor} = (\text{Gear Ratio})^{1/m}$$

Where m = number of separate gear meshes

For example, for a triple reduction gearbox with:

Input RPM = 3594, Assumed Speed Increment = ?
Output RPM = 230, Assumed Interm. #1 RPM = ?
Gear Ratio = 15.625, Assumed Interm. #2 RPM = ?

Gear Ratio = 15.625; and 1/m = 1/3 = .3333
(3 meshes)

Thus, Speed Increment Factor = $(15.625)^{.3333} = 2.50$
Assumed Interm.#1 Speed = 3594/2.50 = 1438 RPM
Assumed Interm.#2 Speed = 1438/2.50 = 575 RPM

Again, when intermediate shaft speeds are confirmed, use these speeds in Case D. And, when the tooth count is confirmed, change spectral band setups immediately back to those specified in Case C of Table III.
Like Case C, the same caution is given on keeping in mind the high frequency limits of the transducer and its mounting. Often, this requires one to stud mount or temporarily epoxy the transducers for these high frequency measurements.

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Case E - Motor Electrical Rotor Bar Pass Frequency Point (single point usually taken on outboard motor bearing):

The specific purpose of this point on each motor is to detect the presence of 1X and 2X rotor bar pass frequencies which often are accompanied by 2X line frequency (7200 CPM) sidebands, and even 2X slip frequency sidebands. The rotor bar pass frequency (RBPF) is equal to the number of rotor bars times motor RPM. High amplitudes at rotor bar pass frequencies suggest rotor bar looseness and/or rotor eccentricity, particularly when these frequencies are accompanied by the 2X line and/or 2X slip frequency sidebands.

Please note that this data is only taken at one point on each motor (normally on the outboard horizontal housing). Notice that the maximum frequency for this point (F_{MAX}) is fixed at 360,000 CPM. Also, note that Band 1 begins at 30,000 CPM and incrementally takes 55,000 CPM steps in each succeeding band up to 360,000 CPM in Band 6 (independent of operating speed for this point which applies to 900 to 3600 RPM motors). Here again, recall that the standard points also taken on this motor (specified using either Case A or B, depending on the bearing type) will evaluate unbalance, misalignment, etc. The number of rotor bars in most all motors is rarely known, but normally ranges between about 35 to 90. Therefore, the F_{MAX} of 360,000 CPM should almost always capture both the first and second harmonic rotor bar pass frequencies, even on 2-pole, nominal 3600 CPM motors. 1600 line spectra are recommended here with 8 to 16 spectrum averages. Since F_{MAX} is so high, even 16 averages of 1600 line spectra should require only about 7 to 10 seconds total. However, due to the high frequency, this might require permanent placement of a disk using a thin layer of high frequency epoxy adhesive in order to provide a dependable measurement mounting. Use of a high strength, rare-earth magnet is recommended in order to provide a good transducer mounting for this important electrical measurement.

Case F - Motor Electrical 12,000 CPM Measurement Point (single point usually taken on inboard motor bearing):

The whole purpose of this single point on each motor is to (1) attempt to separate mechanical and electrical vibration frequencies, particularly in the area of 1X RPM, line frequency (3600 CPM or 60 Hz), and 2X line frequency (7200 CPM or 120 Hz); and (2) to detect the possible presence of pole pass frequency sidebands around running speed harmonics. Very often in predictive maintenance programs, the spectrum will show high vibration at a so-called frequency of 7200 CPM which might suggest electrical problems. However, unless one has the required frequency resolution to separate running speed harmonics from the electrical synchronous frequencies, he cannot truly detect the presence of either a mechanical or an electrical problem, its severity, and certainly not its cause (variable air gap, stator problems, etc.). This is due to the fact that with an F_{MAX} of 40X RPM, he cannot separate, for example, the 3550 RPM operating speed peak from the 3600 CPM line frequency.

Therefore, if one uses 3200 FFT lines of resolution and a 12,000 CPM F_{MAX}, he will likely be able to separate most of these mechanical and electrical frequencies, depending on the motor RPM. Note that 400 FFT lines with a 12,000 CPM F_{MAX} will result in a 30 CPM frequency resolution which means peaks must be at least 45 CPM apart to show two separate frequencies (resolution X 1.5 Hanning Noise Factor). For example, if the speed of a 2-pole motor is approximately 3550 RPM, one would be able to separate running speed from 3600 CPM line frequency and 2X RPM (7100 CPM) from 2X line frequency (7200 CPM) using only 400 lines and a 12,000 CPM F_{MAX}. However, if the motor speed were higher in the range of 3590 to 3595 RPM (5 to 10 CPM slip frequency X 2 poles = 10 to 20 CPM pole pass frequency, it will require 3200 lines of resolution to display both the running speed harmonics and pole pass sidebands (3200 lines with a span of 12,000 CPM will give a frequency resolution of 3.75 CPM and a bandwidth of 5.625 CPM allowing the analyst to see each set of frequencies). Due to the high resolution of 3200 lines and rather small frequency span of 12,000 CPM, this will require 16 seconds for the first average. Therefore, it is recommended that 50% overlap processing be used taking 2 averages for this measurement

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(which will result in 16 seconds for the first average and 8 seconds for the second average, or a total of 24 seconds). However, the end result of this measurement is critical. It alone will allow the analyst to separate mechanical and electrical problems, plus detect potentially serious cracked or broken rotor bars (which he cannot even detect using 400 lines and an F_{MAX} of 40X RPM or so). It will also allow detection of subsynchronous belt defect frequencies on belt-driven machinery.

Case G - Special Machine Types

This section will cover various special machine types which, for some reason, require different spectral band setups from the general machine types covered in Cases A and B in Table III. For example, the primary purpose for specifying bands specifically for centrifugal machines (i.e., pumps and fans) in Types 1 through 4 under Case G is to separate the blade pass frequency band from the bearing defect frequency band. The problem here is that amplitudes which would be acceptable to blade pass frequencies would normally be excessive for a bearing defect frequency. If both the blade pass frequency and bearing frequencies coexisted within the same band, it would be impossible to separate the alarm levels for these two sources. This will be discussed in following sections:

Type 1 - Centrifugal Machines with Known Number of Vanes (or Blades) and Rolling Element Bearings:

If one knows the number of vanes (or blades) in his pump, fan or compressor, it will be possible to set up a separate band to capture blade pass frequency (BPF), allowing a higher alarm for it than that for the bands containing bearing defect frequencies (BPFI, BPFO, etc.). This procedure is illustrated in Table III under Type 1 of Case G. Band 4 will include the fundamental blade pass frequency (BPF) as well as the first running speed sidebands above and below BPF. This band will have an alarm level of 60% of the overall alarm. On the other hand, bearing frequency Bands 3 and 5 on either side of Band 4 will have much lower alarms as seen in Table III. Notice that Band 5 will likely capture not only lower harmonic bearing frequencies, but also harmonic blade pass frequencies which might relate to flow pulsation problems.

Type 2 - Centrifugal Machines with Unknown Number of Vanes (or Blades) and Rolling Element Bearings:

Type 2 will cover rolling element bearing machines when the number of vanes or blades is unknown. In this case, the frequency limits for the probable BPF Band 4 are set to capture what should be blade pass frequency for roughly 60% to 80% of pumps and fans which often have 4 to 6 vanes (or blades). Of course, if this is not the case, Band 4 can be adjusted. In any case, when the number of blades (or vanes) is found, replace the spectral alarm bands shown here with those given in Type 1. Notice in Type 2 that the Band 4 alarm is set at 60% of the overall alarm as in the case of Type 1. Here, the intention is to ensure that if fundamental bearing frequencies do occur in this band, amplitudes will not be allowed to become highly excessive before a potential bearing problem (possibly with frequencies within the same band as blade pass) is detected.

Type 3 - Centrifugal Machines with Known Number of Vanes (or Blades) and Sleeve Bearings:

In this case, the maximum frequency will be set at only 20X RPM or 1.2X Blade Pass (whichever is greater), versus 40 to 100X RPM in the case of a unit with rolling element bearings. Here, Band 5 will capture the blade pass frequency (BPF) as well as 1X RPM sidebands above and below BPF. Band 5 will have an alarm set at 70% of the overall, versus only 30% of the overall alarm in the case of Bands 4 and 6 to its left and right.

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Type 4 - Centrifugal Machines with Unknown Number of Vanes (or Blades) and Sleeve Bearings:

Here, Band 4 will likely capture a large number of blade pass frequencies for pumps and fans since it will range between 4X and 6.5X RPM. Band 6 (ranging from 10X to 20X RPM), will likely capture blade pass frequency harmonics, again relating to flow pulsation problems. Here again, when the number of blades (or vanes) is found, replace these spectral alarm bands with the setup shown in Type 3 for sleeve bearing units.

7.151 EXAMPLES - Specification of Spectral Alarm Bands for Sample Machines:

Please refer to the Sample Machines shown in Figures 2 and 3, and then to the sample spectral alarm band tables which have been worked out for them. Note that the tables themselves are designed for direct input into the predictive maintenance software once the alarm bands have been specified on them. Please note in particular how both the standard frequency and high frequency points are set up for the example with the 2-step gear increaser shown in Figure 3.

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Metric Version Of Table II

TABLE II

TECHNICAL ASSOCIATES OF CHARLOTTE, P.C.

CRITERIA FOR OVERALL CONDITION RATING (RMS OVERALL VELOCITY, mm/sec.)*

1. Assuming Machine Speed = 600 to 60,000 RPM.
2. Assuming Measurements by Accelerometer or Velocity Pickup securely mounted as Close as Possible to Bearing Housing.
3. Assuming Machine Is Not Mounted on Vibration Isolators (for Isolated Machinery, Set Alarm 30% - 50% Higher).
4. Set Motor Alarms the Same as that for the Particular Machine Type unless Otherwise Noted.
5. Set Alarms on Individual External Gearbox Positions about 25% Higher than that for a particular Machine Type.

MACHINE TYPE	GOOD	FAIR	(WARNING) ALARM 1	(FAULT) ALARM 2
COOLING TOWER DRIVES				
Long, Hollow Drive Shaft	0 - 7.0	7.0 - 11.0	11.0	16.0
Close Coupled Belt Drive	0 - 5.0	5.0 - 7.5	7.5	12.0
Close Coupled Direct Drive	0 - 3.5	3.5 - 5.5	5.5	8.0
COMPRESSORS				
Reciprocating	0 - 6.0	6.0 - 9.0	9.0	13.5
Rotary Screw	0 - 5.5	5.5 - 8.0	8.0	12.0
Centrifugal With or W/O External Gearbox	0 - 3.5	3.5 - 5.5	5.5	8.0
Centrifugal - Integral Gear (Axial Meas.)	0 - 3.5	3.5 - 5.5	5.5	8.0
Centrifugal - Integral Gear (Radial Meas.)	0 - 2.5	2.5 - 4.5	4.5	6.5
BLOWERS (FANS)				
Lobe-Type Rotary	0 - 5.5	5.5 - 8.0	8.0	12.0
Belt-Driven Blowers	0 - 5.0	5.0 - 7.5	7.5	11.5
General Direct Drive Fans (with Coupling)	0 - 4.5	4.5 - 6.5	6.5	10.0
Primary Air Fans	0 - 4.5	4.5 - 6.5	6.5	10.0
Large Forced Draft Fans	0 - 3.5	3.5 - 5.5	5.5	8.0
Large induced Draft Fans	0 - 3.0	3.0 - 5.0	5.0	7.0
Shaft-Mounted Integral Fan (Extended Motor Shaft)	0 - 3.0	3.0 - 5.0	5.0	7.0
Vane-Axial Fans	0 - 2.5	2.5 - 4.5	4.5	6.5
MOTOR/GENERATOR SETS				
Belt-Driven	0 - 5.0	5.0 - 7.5	7.5	12.0
Direct Coupled	0 - 3.5	3.5 - 5.5	5.5	8.0
CHILLERS				
Reciprocating	0 - 4.5	4.5 - 7.0	7.0	10.5
Centrifugal (Open-Air) - Motor & Comp. Separate	0 - 3.5	3.5 - 5.5	5.5	8.0
Centrifugal (Hermetic) - Motor & Impellers Inside	0 - 2.5	2.5 - 4.0	4.0	6.0
LARGE TURBINE/GENERATORS				
3600 RPM Turbine/Generators	0 - 3.0	3.0 - 5.0	5.0	7.0
1800 RPM Turbine/Generators	0 - 2.5	2.5 - 4.0	4.0	6.0
CENTRIFUGAL PUMPS				
Vertical Pumps (4m - 6m Height) } Height from Grade to Top Motor Bearing.	0 - 6.0	6.0 - 9.0	9.0	13.5
Vertical Pumps (3m - 4m Height) } May be Necessary to Spec. Lower Alarm	0 - 5.0	5.0 - 7.5	7.5	11.5
Vertical Pumps (2m - 3m Height) } for Lower Motor Bearing & for Upper	0 - 4.0	4.0 - 6.0	6.0	9.5
Vertical Pumps (0m -2m Height) } Pump Bearing (depending on height).	0 - 3.5	3.5 - 5.5	5.5	8.0
General Purpose Horizontal Pump - Direct Coupled	0 - 3.5	3.5 - 5.5	5.5	8.0
Boiler Feed Pumps - Horizontal Orientation	0 - 3.5	3.5 - 5.5	5.5	8.0
Hydraulic Pumps - Horizontal Orientation	0 - 2.0	2.0 - 3.5	3.5	5.5
MACHINE TOOLS				
Motor	0 - 1.2	1.2 - 1.8	1.8	2.7
Gearbox Input	0 - 1.8	1.8 - 2.7	2.7	4.0
Gearbox Output	0 - 1.2	1.2 - 1.8	1.8	2.7
Spindles:				
a. Roughing Operations	0 - .70	.70 - 1.2	1.2	1.8
b. Machine Finishing	0 - .45	.45 - .72	.72	1.1
c. Critical Finishing	0 - .27	.27 - .45	.45	.72

***NOTE:The "ALARM 1" and "ALARM 2" overall levels given above apply only to in-service machinery which has been operating for some time after initial installation and/or overhaul. They do not apply (and are not meant to serve as) Acceptance Criteria for either new or rebuilt machinery.**

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TABLE III RECOMMENDED SPECTRAL ALARM BANDS FOR VARIOUS MACHINE TYPES (*See Notes Below Table)

CASE	ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
A. GENERAL ROLLING ELEMENT BEARING COMPONENTS WITHOUT VANES - **Set F_{MAX} = 50X RPM** (But, if Speed <1700 RPM or if Bearing is Tapered or Spherical Roller Type, See Notes b & c). (i.e., AC & DC Motors, Gearboxes)							
		Gearboxes require second measurement with F_{MAX} and #Lines specified in Cases C and D.					
	BAND LOWER FREQ.	1% F_{MAX}	1.2 X RPM	2.2 X RPM	3.2 X RPM	12.2 X RPM	50% F_{MAX}
	BAND HIGHER FREQ.	1.2 X RPM	2.2 X RPM	3.2 X RPM	12.2 X RPM	50% F_{MAX}	100% F_{MAX}
	BAND ALARM	90% OA ALARM	30% OA ALARM	25% OA ALARM	20% OA ALARM NTE .080 in/sec	15% OA ALARM NTE .060 in/sec	10% OA ALARM NTE .050 in/sec
	DESCRIPTION OF BAND COVERAGE	(Subsynchronous thru 1X RPM)	(1.5 X - 2.0 X RPM)	(2.5 X - 3.0 X RPM)	(Fundamental Bearing Defect Frequencies)	(Lower Harmonic Bearing Frequencies)	(Higher Harmonic Bearing Freqs & Brg. Natl. Freqs.)
B. GENERAL SLEEVE BEARING MACHINES WITHOUT VANES (i.e., AC & DC Motors, Gearboxes) - **Set F_{MAX} = 20X RPM. (For Gearboxes, also see Case C and D).**							
	BAND LOWER FREQ.	0.2 X RPM	0.8 X RPM	1.8 X RPM	2.8 X RPM	3.8 X RPM	10.2 X RPM
	BAND HIGHER FREQ.	0.8 X RPM	1.8 X RPM	2.8 X RPM	3.8 X RPM	10.2 X RPM	100% F_{MAX}
	BAND ALARM	20% OA ALARM NTE .060 in/sec	90% OA ALARM	30% OA ALARM	25% OA ALARM	20% OA ALARM	15% OA ALARM
	DESCRIPTION OF BAND COVERAGE	(Subsynchronous Band)	(1 X - 1.5 X RPM)	(2 X - 2.5 X RPM))	(3 X - 3.5 X RPM)	(4 X - 10 X RPM)	(10.5 X RPM - F_{MAX})
		Gearboxes require second measurement with F_{MAX} and #Lines specified in Cases C and D.					
C. GEARBOX SPECIAL HIGH FREQUENCY POINTS WITH KNOWN NUMBER OF TEETH - **Set F_{MAX} = 3.25X Gear Mesh Frequency (GMF) - (However, see Case C Note Below)**							
	BAND LOWER FREQ.	0.25 X GMF	0.75 X GMF	1.25 X GMF	1.75 X GMF	2.25 X GMF	2.75 X GMF
	BAND HIGHER FREQ.	0.75 X GMF	1.25 X GMF	1.75 X GMF	2.25 X GMF	2.75 X GMF	100% F_{MAX}
	BAND ALARM	25% OA ALARM	70% OA ALARM	25% OA ALARM	50% OA ALARM	25% OA ALARM	40% OA ALARM
	DESCRIPTION OF BAND COVERAGE	(0.5 X GMF & Sidebands)	(GMF & SIDEBANDS)	(1.5 X GMF & SIDEBANDS)	(2 X GMF & SIDEBANDS)	(2.5 X GMF & SIDEBANDS)	(3 X GMF & SIDEBANDS)
D. GEARBOX HIGH FREQUENCY POINTS WITH UNKNOWN NUMBER OF TEETH - **Set F_{MAX} = 200X Shaft RPM at that particular location (However, see Notes a thru d)**							
	BAND LOWER FREQ.	20 X RPM	50 X RPM	80 X RPM	110 X RPM	140 X RPM	170 X RPM
	BAND HIGHER FREQ.	50 X RPM	80 X RPM	110 X RPM	140 X RPM	170 X RPM	100% F_{MAX}
	BAND ALARM	60% OA ALARM	60% OA ALARM	50% OA ALARM	50% OA ALARM	40% OA ALARM	40% OA ALARM
	DESCRIPTION OF BAND COVERAGE	(20X - 50X RPM)	(50X - 80X RPM)	(80X - 110X RPM)	(110X - 140X RPM)	(140X - 170X RPM)	(170X RPM - F_{MAX})

NOTES: a. Obtain Overall Alarm Velocity from Table II entitled "TECHNICAL ASSOCIATES CRITERIA FOR CONDITION RATING" . Also "F_{MAX}" = Maximum Frequency for Spectrum (CPM or Hz).
b. If bearing is a Tapered Roller Bearing or Spherical Roller Bearing, set F_{MAX} = 60 X RPM.
c. If Speed = 1400-1700 RPM, set F_{MAX} = 60 X RPM; if 1100-1399 RPM, set F_{MAX} = 80 X RPM; if 800-1099 RPM, set F_{MAX} = 100 X RPM②; if 600-799 RPM, set F_{MAX} = 120XRPM④.
(To Detect Bearing Natural Frequencies)

NOTE: On Gearboxes, specify these high frequency points checking up to 3.25 X GMF as well as points with normal frequency ranges (as in Case A or B above) on these same points (i.e., **2 separate measurements on same point with 2 different frequency ranges**). Specify F_{MAX} = 3.25 X GMF and locate points close to mesh locations of each set of gears using the specific gear mesh frequency at that particular location: However, remember that amplitudes at high frequencies approaching or exceeding the natural frequency of accelerometer mounting may be questionable (of transducer stud, magnet or probe mounting). In many cases, transducer may require stud or adhesive mounting and measurement of acceleration due to high F_{MAX}. **Use 1600-3200 FFT Lines** in order to allow detection of 1X RPM Sidebands of each meshing gear.

NOTE: a. Specify both high frequency as well as standard frequency measurement points at each location as in Case C. In many cases, transducer may require stud or adhesive mounting and measurement of acceleration due to high F_{MAX}.
b. **Use 3200 FFT Lines** to detect 1X RPM Sidebands of each meshing gear.
c. RPM = Shaft speed at the particular measurement point for which you are specifying spectral bands. If only input and output gearbox speeds are known in a multi-stage gearbox, assume an equal speed increment at each mesh (for example, for a 16:1 ratio, double reduction gearbox, assume approx. a 4.0 speed ratio at each of the mesh points as a starting point - reline when actual speeds are known).

*NOTES: 1. These Spectral Alarm Band Specs apply to "**ALARM 1**" for standard process and utility machinery such as centrifugal pumps, blowers, motors, FD & ID fans, motor/generator sets, refrigeration chillers, centrifugal air compressors, vacuum pumps, boiler feed pumps, gearboxes, etc. "**ALARM 2**" (higher alarm band setting) should have alarm levels 50% greater than "**ALARM 1**".
2. These Specs do not apply to other machine types such as reciprocating, axial screw or lobe-type rotary compressors; diesel engines; gas turbines; large turbine/generators; exciters; rotary blowers; pulverizers, etc. Spectral Bands for these types normally have to be "custom-designed" for each of these particular machines or group of similar machines, often after complete vibration spectra have been captured one or more times on these machines.
3. "NTE" = Not to Exceed; "OA ALARM" = Overall Alarm Specified in Table II.
4. **Frequency Resolution = 800 Lines** unless otherwise instructed; **If F_{MAX} > 100X RPM**, increase resolution to **1600 lines** unless otherwise instructed; **Use 50% Overlap Averaging.**

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TABLE III

RECOMMENDED SPECTRAL ALARM BANDS FOR VARIOUS MACHINE TYPES (*See Notes Below Table)

CASE	ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
E. AC INDUCTION MOTOR ELECTRICAL ROTOR BAR PASS FREQUENCY POINT (MEASURE ON MOTOR OUTBOARD BEARING HORIZONTAL) - Set F_{MAX} = 360,000 CPM (See Notes a thru d).							
	BAND LOWER FREQ.	30,000 CPM	85,000 CPM	140,000 CPM	195,000 CPM	250,000 CPM	305,000 CPM
	BAND HIGHER FREQ.	85,000 CPM	140,000 CPM	195,000 CPM	250,000 CPM	305,000 CPM	360,000 CPM
	BAND ALARM	.060 in/sec.	.055 in/sec.	.050 in/sec.	.040 in/sec	.030 in/sec	.030 in/sec
	DESCRIPTION OF BAND COVERAGE	(30K - 85K)	(85K - 140K)	(140K - 195K)	(195K - 250K)	(250K - 305K)	(305K - 360K)
NOTES: a. Purpose of this Point on each motor is to detect possible 1X and 2X Rotor Bar Pass Frequencies (RBPF) which may be accompanied by 2X line frequency sidebands (normally at 7200 CPM). b. Case E applies to motors ranging in speeds from nominal 900 to 3600 RPM. c. Ensure that transducer is well mounted on the motor housing due to the rather high F_{MAX} . Normally, it should be taken in the horizontal direction on the outboard motor bearing housing. d. Should use **1600 FFT Lines** & 8 Averages to clearly identify possible Rotor Bar Pass Frequencies as well as 2X Line Frequency and/or 1X RPM Sidebands.								
F. AC INDUCTION MOTOR ELECTRICAL MEASUREMENT POINT ON MOTOR INBOARD BEARING HORIZONTAL - Set F_{MAX} = 24,000 CPM (For nominal 600 - 3600 RPM motors).							
	BAND LOWER FREQ.	0.8 X Line Freq.	3.8 X Line Freq.	NOT REQUIRED	NOT REQUIRED	NOT REQUIRED	NOT REQUIRED
	BAND HIGHER FREQ.	1.2 X Line Freq.	4.2 X Line Freq.				
	BAND ALARM	.07 in/sec	.035 in/sec				
	DESCRIPTION OF BAND COVERAGE	(2 F_{LINE} ± Pole Pass Freq. Sidebands)	(4 F_{LINE} ± Pole Pass Freq. Sidebands)				
NOTES: a. Purpose of this Point is to attempt to separate Mechanical and Electrical vibration peaks, particularly in vicinity of 2X Line Frequency and 4X Line Frequency. b. Should use **6400 FFT Lines** and 2 Averages to clearly separate Mechanical & Electrical Frequencies and to detect Pole Pass Freq. Sidebands (F_P = Slip Freq. X #Poles). c. This measurement is **Required** on Each Motor to Detect Pole Pass Freq. Sidebands surrounding either 1X RPM Harmonics and/or 2X Line Frequency.								
G. CENTRIFUGAL COMPRESSORS, BLOWERS AND PUMPS **TYPE 1. DRIVEN CENTRIFUGAL COMPONENT WITH KNOWN NUMBER OF VANES (OR BLADES) AND ROLLING ELEMENT BEARINGS - Set F_{MAX} = 50X RPM (See Notes a thru c).**							
	BAND LOWER FREQ.	0.4 X RPM	1.2 X RPM	2.2 X RPM	BPF - 1.2 X RPM	BPF + 1.2 X RPM	50% F_{MAX}
	BAND HIGHER FREQ.	1.2 X RPM	2.2 X RPM	(BPF - 1.2 X RPM)	BPF + 1.2 X RPM	50% F_{MAX}	100% F_{MAX}
	BAND ALARM	90% OA ALARM	30% OA ALARM	35% OA ALARM NTE .120 in/sec.	60% OA ALARM NTE .240 in/sec.	35% OA ALARM NTE .120 in/sec.	20% OA ALARM NTE .070 in/sec.
	DESCRIPTION OF BAND COVERAGE	(Subsynchronous thru 1 X RPM)	(1.5 -2.0 X RPM)	(2.5 X - Funda. Bearing Defect Frequencies)	(BPF ± 1 X Sidebands) BPF = Blade Pass Freq.	(Lower Harmonic Bearing Freqs. & BPF Harmonics)	(Higher Harmonic Bearing Freqs. & Brg. Natl. Freqs.)
NOTES: a. Specify separate set of spectral alarm bands for Driver (as shown in Case A or B for motor, gearbox, etc.) where BPF is not involved. b. "BPF" = Blade Pass Frequency = No. of Blades (or Vanes) X RPM. If bearing is a Tapered Roller Bearing or Spherical Roller Bearing, Set F_{MAX} =60X RPM unless speed< 1700 RPM (see note b in Type 1). c. If Speed = 1400-1700 RPM, set F_{MAX} = 60X RPM; if 1100-1399 RPM, set F_{MAX} = 80X RPM; if 800-1099 RPM, set F_{MAX} = 100 X RPM④; if 600-799 RPM, set F_{MAX} = 120X RPM④.							
TYPE 2A. CENTRIFUGAL PUMPS WITH UNKNOWN NUMBER OF VANES AND ROLLING ELEMENT BEARINGS -Set F_{MAX} = 50X RPM (See Notes a thru c).							
	BAND LOWER FREQ.	1% F_{MAX}	1.2 X RPM	2.2 X RPM	3.2 X RPM	6.8 X RPM	50% F_{MAX}
	BAND HIGHER FREQ.	1.2 X RPM	2.2 X RPM	3.2 X RPM	6.8 X RPM	50% F_{MAX}	100% F_{MAX}
	BAND ALARM	90% OA ALARM	30% OA ALARM	25% OA ALARM	60% OA ALARM NTE .185 in/sec.	30% OA ALARM NTE .120 in/sec.	20% OA ALARM NTE .070 in/sec.
	DESCRIPTION OF BAND COVERAGE	(Subsynchronous thru 1 X RPM)	(1.5 -2.0 X RPM)	(2.5-3.0 X RPM)	(Possible BPF for Pumps)	(Lower Harmonic Bearing Freq. & BPF Harmonics)	(Higher Harmonic Bearing Freqs. & Brg. Natl. Freqs.)
NOTES: a. If bearing is a Tapered roller Bearing or Spherical Roller Bearing, Set F_{MAX} = 60 X RPM unless Speed < 1700 RPM. b. If Speed = 1400-1700 RPM, set F_{MAX} = 60X RPM; if 1100-1399 RPM, set F_{MAX} = 80X RPM; if 800-1099 RPM, set F_{MAX} = 100 X RPM④; if 600-799 RPM, set F_{MAX} = 120X RPM④. c. When Number of Vanes is confirmed, replace these Spectral Alarm Bands **IMMEDIATELY** with setup shown in Case G, Type 1.							

*NOTE: 1. These Spectral Alarm Band Specs apply to **"ALARM 1"** for standard process and utility machinery such as centrifugal pumps, blowers, motors, FD & ID fans, motor/generator sets, refrigeration chillers, centrifugal air compressors, vacuum pumps, boiler feed pumps, gearboxes, etc. **"ALARM 2"** (higher alarm band setting) should have alarm levels 50% greater than **"ALARM 1"**.

2. These Specs do not apply to other machine types such as reciprocating, axial screw or lobe-type rotary compressors; diesel engines; gas turbines; large turbine/generators; exciters; rotary blowers; pulverizers, etc. Spectral Bands for these types normally have to be "custom-designed" for each of these particular machines or group of similar machines, often after complete vibration spectra have been captured one or more times on these machines.

3. "NTE" = Not to Exceed; "OA ALARM" = Overall Alarm Specified in Table II.

4. **Frequency Resolution = 800 Lines** unless otherwise instructed; **If F_{MAX} > 100X RPM**, increase resolution to **1600 lines** unless otherwise instructed; **Use 50% Overlap Averaging**

TABLE III

RECOMMENDED SPECTRAL ALARM BANDS FOR VARIOUS MACHINE TYPES (*See Notes Below Table)

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CASE	ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
TYPE 2B. CENTRIFUGAL BLOWERS & COMPRESSORS WITH UNKNOWN NUMBER OF BLADES AND ROLLING ELEMENT BEARINGS - Set F_{MAX} = 50X RPM (See Notes a thru c).							
	BAND LOWER FREQ.	1% F_{MAX}	1.2 X RPM	3.2 X RPM	7.8 X RPM	12.8 X RPM	50% F_{MAX}
	BAND HIGHER FREQ.	1.2 X RPM	3.2 X RPM	7.8 X RPM	12.8 X RPM	50% F_{MAX}	100% F_{MAX}
	BAND ALARM	90% OA ALARM	35% OA ALARM	30% OA ALARM	40% OA ALARM NTE .100 in/sec.	20% OA ALARM NTE .080 in/sec	15% OA ALARM NTE .060 in/sec
	DESCRIPTION OF BAND COVERAGE	(Subsynchronous thru 1X RPM)	(1.5 - 3.0X RPM)	(3.5 - 7.5 X RPM)	(Possible BPF for Fans & Compressors)	(Lower Harmonic Bearing Freq. & BPF Harmonics)	(Higher Harmonic Beaing Freq. & BPF Harmonics)
NOTES: a. If bearing is a Tapered Roller Bearing or Spherical Roller Bearing, Set F_{MAX} = 60 X RPM unless Speed < 1700 RPM. (see Note b). b. If Speed = 1400-1700 RPM, set F_{MAX} = 60X RPM; if 1100-1399 RPM, set F_{MAX} = 80X RPM; if 800-1099 RPM, set F_{MAX} = 100 X RPM④; if 600-799 RPM, set F_{MAX} = 120X RPM④. c. When Number of Blades is confirmed, replace these Spectral Alarm Bands **IMMEDIATELY** with setup shown in Case G, Type 1.								
TYPE 3 . DRIVEN CENTRIFUGAL COMPONENT WITH KNOWN NUMBER OF VANES (OR BLADES) AND WITH SLEEVE BEARINGS - Set F_{MAX} =20X RPM or 1.2X BPF, Whichever is Greater .							
	BAND LOWER FREQ.	0.3 X RPM③	0.8 X RPM	1.8 X RPM	3.8 X RPM	(BPF - 1.2 X RPM)	(BPF + 1.2 X RPM)
	BAND HIGHER FREQ.	0.8 X RPM	1.8 X RPM	3.8 X RPM	(BPF - 1.2 X RPM)	(BPF + 1.2 X RPM)	100% F_{MAX}
	BAND ALARM	20% OA ALARM NTE .06 in/sec	90% OA ALARM	40% OA ALARM	30% OA ALARM	70% OA ALARM NTE .240 in/sec	30% OA ALARM NTE .100 in/sec
	DESCRIPTION OF BAND COVERAGE	(Subsynchronous Band)	(1X - 1.5X RPM)	(2X - 3.5 X RPM)	(4X - Lower RPM Harmonics)	(BPF ± 1X Sidebands)	(Higher RPM Harmonics and BPF Harmonics)
NOTES: a. Specify separate set of spectral alarm bands for Driver (as shown in Case A or B for motor, gearbox, etc.) where BPF is not involved. b. "BPF" = Blade Pass Frequency = No. of Blades (or Vanes) X RPM. c. If Number of Vanes >25, set Band 1 Lower Frequency = 0.4 X RPM.							
TYPE 4A. CENTRIFUGAL, PUMPS EQUIPPED WITH UNKNOWN NUMBER OF VANES AND WITH SLEEVE BEARINGS - Set F_{MAX} = 20X RPM (See Notes a and b).							
	BAND LOWER FREQ.	0.3 X RPM	0.8 X RPM	1.8 X RPM	3.8 X RPM	7.2 X RPM	9.8 X RPM
	BAND HIGHER FREQ.	0.8 X RPM	1.8 X RPM	3.8 X RPM	7.2 X RPM	9.8 X RPM	100% F_{MAX}
	BAND ALARM	20% OA ALARM NTE .06 in/sec.	90% OA ALARM	40% OA ALARM	70% OA ALARM NTE .240 in/sec.	25% OA ALARM	35% OA ALARM NTE .120 in/sec.
	DESCRIPTION OF BAND COVERAGE	(Subsynchronous Band)	(1X - 1.5X RPM)	(2X - 3.5X RPM)	(Possible BPF for Pumps)	(8X - 9.5X RPM)	(10X RPM - F_{MAX})
NOTES: a. Specify separate set of spectral alarm bands for Driver (as shown in Case A or B for motor, gearbox, etc.) where BPF is not involved. b. When Number of Vanes is found, replace these Spectral Alarm Bands **IMMEDIATELY** with setup shown in Case G, Type 3.							
TYPE 4B. CENTRIFUGAL BLOWERS & COMPRESSORS WITH UNKNOWN NUMBER OF BLADES AND WITH SLEEVE BEARINGS -Set F_{MAX} = 30X RPM (See Notes a and b).							
	BAND LOWER FREQ.	0.3 X RPM	0.8 X RPM	1.8 X RPM	3.2 X RPM	7.2 X RPM	12.8 X RPM
	BAND HIGHER FREQ.	0.8 X RPM	1.8 X RPM	3.2 X RPM	7.2 X RPM	12.8 X RPM	100% F_{MAX}
	BAND ALARM	20% OA ALARM NTE .06 in/sec.	90% OA ALARM	40% OA ALARM	30% OA ALARM	40% OA ALARM NTE .100 in/sec.	20% OA ALARM NTE .060 in/sec.
	DESCRIPTION OF BAND COVERAGE	(Subsynchronous Band)	(1X - 1.5X RPM)	(2X - 3X RPM)	(3.5 - 7.5X RPM)	(Possible BPF for Fans and Compressors)	(13X RPM - F_{MAX})
NOTES: a. Specify separate set of spectral alarm bands for Driver (as shown in Case A or B for motor, gearbox, etc.) where BPF is not involved. b. When Number of Blades is found, replace these Spectral Alarm Bands **IMMEDIATELY** with setup shown in Case G, Type 3.							

*NOTE: 1. These Spectral Alarm Band Specs apply to "**ALARM 1**" for standard process and utility machinery such as centrifugal pumps, blowers, motors, FD & ID fans, motor/generator sets, refrigeration chillers, centrifugal air compressors, vacuum pumps, boiler feed pumps, gearboxes, etc. "**ALARM 2**" (higher alarm band setting) should have alarm levels 50% greater than "**ALARM 1**".

2. These Specs do not apply to other machine types such as reciprocating, axial screw or lobe-type rotary compressors; diesel engines; gas turbines; large turbine/generators; exciters; rotary blowers; pulverizers, etc. Spectral Bands for these types normally have to be "custom-designed" for each of these particular machines or group of similar machines, often after complete vibration spectra have been captured one or more times on these machines.

3. "NTE" = Not to Exceed; "OA ALARM" = Overall Alarm Specified in Table II.

4. **Frequency Resolution = 800 Lines** unless otherwise instructed; **If F_{MAX} > 100X RPM**, increase resolution to **1600 lines** unless otherwise instructed; **Use 50% Overlap Averaging.**

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TABLE III

RECOMMENDED SPECTRAL ALARM BANDS FOR VARIOUS MACHINE TYPES (*See Notes Below Table)

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CASE	ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
H. DC MOTORS: FULL-WAVE AND HALF-WAVE SCR CONTROLLED RECTIFIER (Special Measurements in Addition to Standard Spectral Setups in Cases A & B).							
TYPE 1. FULL-WAVE RECTIFIED MOTORS (MEASURE ON COMMUTATOR-END BEARING HORIZONTAL)- Set F_{MAX} = 24,000 CPM (See Notes a thru c).							
	BAND LOWER FREQ.	3300 CPM	6900 CPM	10,500 CPM	14,100 CPM	17,700 CPM	21,600 - 1.2 X RPM
	BAND HIGHER FREQ.	3900 CPM	7500 CPM	11,100 CPM	14,700 CPM	18,300 CPM	21,600 + 1.2 X RPM
	BAND ALARM	.02 in/sec	.02 in/sec	.02 in/sec	.02 in/sec	.02 in/sec	.08 in/sec
	DESCRIPTION OF BAND COVERAGE	(1 X Line Freq., F_L)	(2 X Line Freq.)	(3 X Line Freq.)	(4 X Line Freq.)	(5 X Line Freq.)	(SCR Firing Freq. ± 1 X RPM Sidebands)
NOTES: a. Purpose of this Special Measurement Point is to Detect Problems with Armature Windings, Commutators, SCR's, Firing Cards, Comparitor Cards, Fuses, etc. b. Must use 6400 Lines and 2 Averages to Detect Δ RPM and 1 X RPM Sidebands around the SCR Firing Frequency; Use 50% Overlap Averaging. c. Assumes Line Frequency = 60 Hz (3600 CPM); Adjust Frequency Span of Each Band if Line Freq. Differs from 60 Hz.							
TYPE 2 . HALF-WAVE RECTIFIED MOTORS (MEASURE ON COMMUTATOR-END BEARING HORIZONTAL) - Set F_{MAX} =24,000 CPM (See Notes a thru c).							
	BAND LOWER FREQ.	3300 CPM	6900 CPM	10,800 - 1.2 X RPM	14,100 CPM	17,700 CPM	21,600 - 1.2 X RPM
	BAND HIGHER FREQ.	3900 CPM	7500 CPM	10,800 + 1.2 X RPM	14,700 CPM	18,300 CPM	21,600 + 1.2 X RPM
	BAND ALARM	.02 in/sec	.02 in/sec	.08 in/sec	.015 in/sec	.015 in/sec	.04 in/sec
	DESCRIPTION OF BAND COVERAGE	(1 X Line Freq., F_L)	(2 X Line Freq.)	(SCR Firing Freq. ± 1 X RPM Sidebands)	(4 X Line Freq.)	(5 X Line Freq.)	(2 X SCR Firing Freq. ± 1 X RPM Sidebands)
NOTES: a. Purpose of this Special Measurement Point is to Detect Problems with Armature Windings, Commutators, SCR's, Firing Cards, Comparitor Cards, Fuses, etc. b. Must use 6400 Lines and 2 Averages to Detect Δ RPM and 1 X RPM Sidebands around the SCR Firing Frequenc;y; Use 50% Overlap Averaging. c. Assumes Line Frequency = 60 Hz (3600 CPM); Adjust Frequency Span of Each Band if Line Freq. Differs from 60 Hz.							
TYPE 3. SPECIAL POINT FOR ELECTRICALLY - INDUCED FLUTING ON FULL-WAVE OR HALF-WAVE RECTIFIED DC MOTORS - SET F_{MAX} = 180,000 CPM (See Notes a thru e).							
	BAND LOWER FREQ.	(SCR - 2.2 X RPM)	(2 X SCR - 2.2 X RPM)	(3 X SCR - 2.2 X RPM)	(4 X SCR - 2.2 X RPM)	(5 X SCR - 2.2 X RPM)	120,000 CPM
	BAND HIGHER FREQ.	(SCR + 2.2 X RPM)	(2 X SCR + 2.2 X RPM)	(3 X SCR + 2.2 X RPM)	(4 X SCR + 2.2 X RPM)	120,000 CPM	180,000 CPM
	BAND ALARM	.08 in/sec	.04 in/sec	.025 in/sec	.020 in/sec	.015 in/sec	.015 in/sec
	DESCRIPTION OF BAND COVERAGE	(SCR Firing Freq. ± 2 X RPM Sidebands)	(2 X SCR Firing Freq. ± 2 X RPM Sidebands)	(3 X SCR Firing Freq. ± 2 X RPM Sidebands)	(4 X SCR Firing Freq. ± 2 X RPM Sidebands)	(BPFO or BPFI Sidebands Around Carrier Freq.)	(BPFO or BPFI Sidebands Around Carrier Freq.)
NOTES: a. Primary Purpose of this Special Measurement is to Detect Fluting normally indicated by BPFO and/or BPFI Sidebands around Carrier Frequency which Ranges from 100,000 to 150,000 CPM. b. SCR Firing Freq. for Half-Wave Rectified DC Motor = 10,800 CPM (180 Hz); but 21,600 CPM (360 Hz) for Full-Wave Rectified DC Motor (with a line frequency (F_L) of 60Hz). c. Must use 3200 lines & 8 averages with F_{MAX} of 180,000 CPM in order to detect electrically induced fluting which is indicated by Spectrum Described in Note a. d. Should take this measurement on Both the Outboard and Inboard Motor Bearings since fluting can occur on either bearing. Use 50% Overlap Averaging. e. Take data in Horizontal Direction on Both bearings and ensure tranducer is well mounted due to the rather high F_{MAX}.							

*NOTE: 1. These Spectral Alarm Band Specs apply to "**ALARM 1**" for standard process and utility machinery such as centrifugal pumps, blowers, motors, FD & ID fans, motor/generator sets, refrigeration chillers, centrifugal air compressors, vacuum pumps, boiler feed pumps, gearboxes, etc. "**ALARM 2**" (higher alarm band setting) should have alarm levels 50% greater than "**ALARM 1**".

2. These Specs do not apply to other machine types such as reciprocating, axial screw or lobe-type rotary compressors; diesel engines; gas turbines; large turbine/generators; exciters; rotary blowers; pulverizers, etc. Spectral Bands for these types normally have to be "custom-designed" for each of these particular machines or group of similar machines, often after complete vibration spectra have been captured one or more times on these machines.

3. "NTE" = Not to Exceed; "OA ALARM" = Overall Alarm Specified in Table II.

4. **Frequency Resolution = 800 Lines** unless otherwise instructed; **If F_{MAX} > 100X RPM**, increase resolution to **1600 lines** unless otherwise instructed; **Use 50% Overlap Averaging.**

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TABLE III

RECOMMENDED SPECTRAL ALARM BANDS FOR VARIOUS MACHINE TYPES (*See Notes Below Table)

CASE	ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
I. MACHINE TOOL SPINDLES (see Table II and Text Section I) - Set F_{MAX} = 50X RPM (But, if speed < 1700 RPM or if Bearing is Tapered or Spherical Roller Type, See Notes a thru c).							
	BAND LOWER FREQ.	0.3 X RPM	0.8 X RPM	1.2 X RPM	2.2 X RPM	3.8 X RPM	50% F_{MAX}
	BAND HIGHER FREQ.	0.8 X RPM	1.2 X RPM	2.2 X RPM	3.8 X RPM	50% F_{MAX}	100% F_{MAX}
	BAND ALARMS:						
	Angular Contact	30% OA ALARM	60% OA ALARM	40% OA ALARM	40% OA ALARM	20% OA ALARM	10% OA ALARM
	Tapered Roller	30% OA ALARM	60% OA ALARM	40% OA ALARM	40% OA ALARM	30% OA ALARM	15% OA ALARM
	DESCRIPTION OF BAND COVERAGE	(FTF & Subsynch.)	(1X RPM)	(1.5X - 2.0X RPM)	(3X RPM & BSF)	(BSF, BPFO, BPFI & Harmonics)	(Bearing/Housing Natural Freqs.)

NOTES: a. Obtain Overall Alarm Velocity from Table II entitled "TECHNICAL ASSOCIATES CRITERIA FOR CONDITION RATING" . Also "F_{MAX}" = Maximum Frequency for Spectrum (CPM or Hz).

c. If Speed = 1400-1700 RPM, set F_{MAX} = 60 X RPM; if 1100-1399 RPM, set F_{MAX} = 80 X RPM; if 800-1099 RPM, set F_{MAX} = 100 X RPM[A]; if 600-799 RPM, set F_{MAX} = 120XRPM[D].
(To Detect Bearing Natural Frequencies) Gearboxes require second measurement with F_{MAX} and #Lines specified in Cases C and D.

d. See Table IIIB for specific maximum allowable amplitudes for discrete bearing defect frequencies and harmonics.

TABLE IIIB. MAXIMUM ALLOWABLE AMPLITUDE AT BPFO, BPFI, BSF OR ANY BEARING FREQUENCY HARMONIC (Multiple) ON MACHINE TOOL SPINDLES (See Note below)

SPINDLE TYPE	TEXT FIG. NO.	MAX. BEARING FREQ. AMPLITUDE	
		ANGULAR CONTACT	CYLIND./ TAPERED
Precision Box Spindle	Fig. 2	.0050 in/sec	.0075 in/sec
Precision Cartridge-Type Spindle	Fig. 3	.0050 in/sec	.0075 in/sec
Precision Multi-Gear Head Spindle	Fig. 4	.0050 in/sec	N/A
Precision Multi-Gear Head Gearbox	Fig. 4	.0075 in/sec	.0075 in/sec
Precision Grinding Wheel Head	Fig. 5	.0050 in/sec	.0075 in/sec
Precision Integral-Motor Spindle	Fig. 6	.0050 in/sec	.0050 in/sec

NOTE: Information in Case I is derived from Technical Associates' ***"Vibration Acceptance Criteria for New and Rebuilt Machinery"*** (Ref. 40)

TABLE IIIC. MAXIMUM BAND LIMITED OVERALL PEAK ACCELERATION FOR MACHINE TOOL SPINDLES (See Note below)

BEARING TYPE	SPINDLE SPEED	FREQUENCY BAND*.	PEAK ACCEL.
Angular Contact	Any Speed	600 - 600,000 CPM	0.50g
Tapered Roller	< 1000 RPM	600 - 600,000 CPM	0.75g
Tapered Roller	> 1000 RPM	600 - 600,000 CPM	1.25g

*NOTE: Great care must be taken in using a proper transducer and in mounting the transducer since measurements are required up to these high frequencies.

*NOTE: 1. These Spectral Alarm Band Specs apply to **"ALARM 1"** for standard process and utility machinery such as centrifugal pumps, blowers, motors, FD & ID fans, motor/generator sets, refrigeration chillers, centrifugal air compressors, vacuum pumps, boiler feed pumps, gearboxes, etc. **"ALARM 2"** (higher alarm band setting) should have alarm levels 50% greater than **"ALARM 1"**.

2. These Specs do not apply to other machine types such as reciprocating, axial screw or lobe-type rotary compressors; diesel engines; gas turbines; large turbine/generators; exciters; rotary blowers; pulverizers, etc. Spectral Bands for these types normally have to be "custom-designed" for each of these particular machines or group of similar machines, often after complete vibration spectra have been captured one or more times on these machines.

3. "NTE" = Not to Exceed; "OA ALARM" = Overall Alarm Specified in Table II.

4. Frequency Resolution = 800 Lines unless otherwise instructed. If F_{MAX} > 100X RPM, increase resolution to 1600 lines unless otherwise instructed. Use 50% Overlap Averaging.

4. **Frequency Resolution = 800 Lines** unless otherwise instructed; **If F_{MAX} > 100X RPM**, increase resolution to **1600 lines** unless otherwise instructed; **Use 50% Overlap Averaging.**

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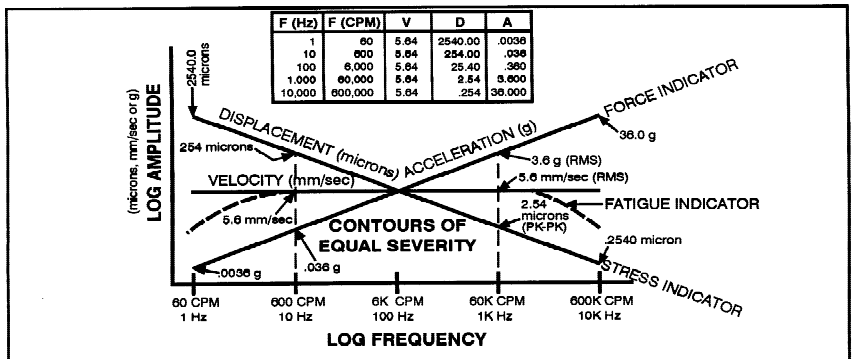
FIGURE 2
SPECTRAL ALARM BAND SPECS ON A HORIZONTAL PUMP


F (Hz) | F (CPM) | V | D | A
1 | 60 | 5.64 | 2540.00 | .0036
10 | 600 | 5.64 | 254.00 | .036
100 | 6,000 | 5.64 | 25.40 | .360
1,000 | 60,000 | 5.64 | 2.54 | 3.600
10,000 | 600,000 | 5.64 | .254 | 36.000
LOG AMPLITUDE
(microns, mm/sec or g)
2540.0 microns
DISPLACEMENT (microns)
254 microns
VELOCITY (mm/sec)
5.6 mm/sec
ACCELERATION (g)
FORCE INDICATOR
36.0 g
3.6 g (RMS)
5.6 mm/sec (RMS)
2.54 microns (PK-PK)
FATIGUE INDICATOR
CONTOURS OF EQUAL SEVERITY
.2540 micron
.036 g
.0036 g
STRESS INDICATOR
60 CPM 1 Hz
600 CPM 10 Hz
6K CPM 100 Hz
60K CPM 1K Hz
600K CPM 10K Hz
LOG FREQUENCY

COMPARISON OF VIBRATION DISPLACEMENT, VELOCITY & ACCELERATION SEVERITY

$V = \frac{D\,F}{27{,}010}$ $\quad D = \frac{27{,}010\,V}{F}$	$A = \frac{D\,F^2}{2.53 \times 10^9}$ $\quad D = \frac{2.53 \times 10^9\,A}{F^2}$
$V = \frac{93{,}580\,A}{F}$ $\quad A = \frac{V\,F}{93{,}580}$	Where: A = RMS Acceleration (g) V = RMS Velocity (mm/sec) D = Peak-Peak Displacement (microns) F = Frequency (CPM)

EXAMPLE 1

TECHNICAL ASSOCIATES OF CHARLOTTE P.C.
(704) 333-9011

BY: JEB
DATE: 5/12/93

CLIENT NAME: XYZ CORP.
MACHINE NAME: HORIZ. PUMP 2

SPECTRAL ALARM BAND COMPUTER ENTRY FORM

POSITION	UNIT	ALARM	RPM	BRGS	F_{MAX}	PARAMETER	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
1A, 1H, 1V 2A, 2H, 2V **(Motor Points As Per Case A)** **(Rotor Bar Pass Freq. Point From Case E)**	in/sec	.300	1780	Ball	72K	LOWER FREQ. HIGHER FREQ. BAND ALARM	720 2160 .270	2160 3960 .120	3960 5760 .090	5760 21,960 .075	21,960 36,000 .060	36,000 72,000 .045
1 BAR **(Motor Points As Per Case A)**	in/sec	.300	1780	Ball	360K	LOWER FREQ. HIGHER FREQ. BAND ALARM	30,000 85,000 .060	85,000 140,000 .060	140,000 195,000 .050	195,000 250,000 .045	250,000 305,000 .040	305,000 360,000 .035
							NO ALARM BANDS FOR SPIKE ENERGY SPECTRA NOW					
2 EL **(Pump Points As Per Case G, Type 1 with 6 Vanes)**	in/sec	.300	1780	Ball	12K	LOWER FREQ. HIGHER FREQ. BAND ALARM	240 2000 .270	2000 4000 .120	4000 6000 .120	6000 8000 .105	8,000 10,000 .090	10,000 12,000 .075
1H, 2H, 3H, 4H	gSE	.700	1780	Ball	____	LOWER FREQ. HIGHER FREQ. BAND ALARM						

OTHER MACHINES WHICH WILL HAVE DUPLICATE SPECTRAL ALARM BANDS:			
HORIZ PUMP 3	HORIZ PUMP 16		
HORIZ PUMP 5			
HORIZ PUMP 7			
HORIZ PUMP 10			
HORIZ PUMP 12			

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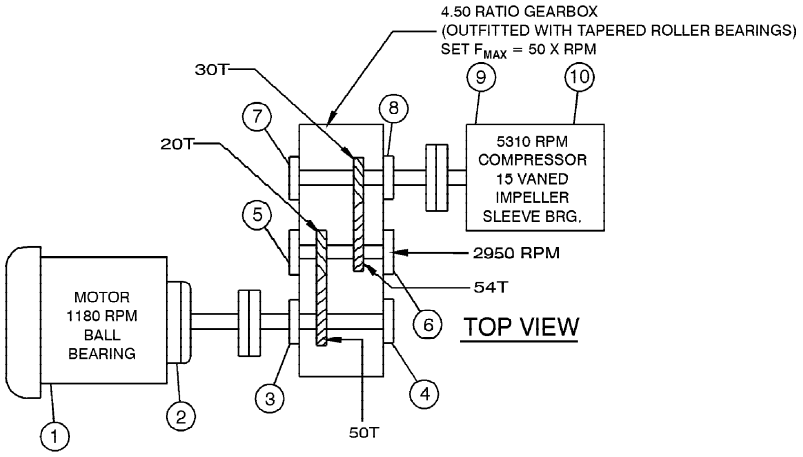
FIGURE 3
EXAMPLE #2- COMPRESSOR DRIVEN BY 2-STAGE GEAR BOX



GMF_1 = (50T)(1180 RPM) = (20T)(2950 RPM)

GMF_1 = 59,000 CPM

POS. 3HI F_{MAX} = GMF X 3.25
= 191,750 CPM

USE POS 3HI F_{MAX} = 200,00 CPM

GMF_2 = 54T X 2950 RPM = (30T)(5310 RPM)

GMF_2 = 159,300 CPM

POS. 6HI F_{MAX} = 3.25 X 159,300
= 517,725 CPM

USE POS 6HI F_{MAX} = 540,00 CPM

(Must Measure Acceleration Here)

EXAMPLE 2

TECHNICAL ASSOCIATES OF CHARLOTTE P.C. (704) 333-9011

BY: JEB
DATE: 5/12/93

MACHINE NAME: COMPRESSOR 2

SPECTRAL ALARM BAND COMPUTER ENTRY FORM

POSITION	UNIT	ALARM	RPM	BRGS		PARAMETER	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
1A, 1H, 1V 2A, 2H, 2V	in/sec	.300	1180	Ball	72K	LOWER FREQ. HIGHER FREQ. BAND ALARM	600 1440 .270	1440 2640 .120	2640 3840 .090	3840 14,640 .075	14,640 36,000 .060	36,000 72,000 .045
1 BAR	in/sec	.300	1180	Ball	360K	LOWER FREQ. HIGHER FREQ. BAND ALARM	30,000 85,000 .060	85,000 140,000 .060	140,000 195,000 .050	195,000 250,000 .045	250,000 305,000 .040	305,000 360,000 .035
2 EL	in/sec	.300	1180	Ball	12K	LOWER FREQ. HIGHER FREQ. BAND ALARM	240 2000 .270	2000 4000 .120	4000 6000 .120	6000 8000 .105	8,000 10,000 .090	10,000 12,000 .075
1H, 2H, 3H, 4H	gSE	.500	1180	Ball	____	LOWER FREQ. HIGHER FREQ. BAND ALARM	NO ALARM BANDS FOR SPIKE ENERGY SPECTRA NOW					
3A, 3H, 3V 4A, 4H, 4V	in/sec	.375	1180	Taper Roller	72K	LOWER FREQ. HIGHER FREQ. BAND ALARM	600 1440 .340	1440 2640 .150	2640 3840 .115	3,840 14,640 .095	14,640 36,000 .075	36,000 72,000 .055
3HI - A&H 5HI - A&H	in/sec	.375	1180	Taper Roller	200K	LOWER FREQ. HIGHER FREQ. BAND ALARM	14,750 44,250 .340	44,250 73,750 .265	73,750 103,250 .095	103,250 132,750 .190	132,750 162,250 .095	162,250 200,000 .150
5A, 5H, 5V 6A, 6H, 6V	in/sec	.375	2950	Taper Roller	144K	LOWER FREQ. HIGHER FREQ. BAND	1440 3540 .340	3540 6490 .150	6490 9440 .115	9440 36,000 .095	36,000 72,000 .075	

***Positions 6HI and 8HI must be Stud or Adhesive Mounted for these High Freq. Acceleration Measurements (to 540,000 CPM).**

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7.16 PERIODIC REEVALUATION OF SPECTRAL ALARM BAND SETUPS ON EACH FAMILY OF MACHINES

Table III gives the procedure on how to specify spectral alarm bands for a number of machine types and bearing configurations. It is intended for use when either originally setting up a PMP database before data is taken, or on the other hand, is intended as a starting point for those who have already taken large numbers of spectral measurements, but have never inserted alarm bands. For those who have specified bands, it serves as a reference to which these setups can be compared.

After baselines and perhaps after several additional surveys are completed with these bands in place, the user should reexamine each of the spectral alarm band setups on each family of machines. That is, he should identify certain families, or groupings of similar machines and evaluate how such bands are working, preferably using statistical analysis (24, 32, 33, 37). He should also evaluate how well the overall alarm levels he has specified for each machine point are functioning. He should ask such questions as:

1. Are the specified alarm bands detecting those problems which are occurring in this machine family - or are they catching some, but missing others?

2. If they are missing some problems, how should they be adjusted in order to better detect them?

3. On the other hand, are these alarm bands causing numerous “false alarms”? If this is allowed to continue, both those in the plant as well as analysts in the program will begin to pay little attention to so-called alarm conditions, even when they actually do detect potentially serious problems.

4. If alarm bands are causing false alarms, how can they be carefully adjusted to noticeably reduce the probability of sounding false alarms, but at the same time, not to the point where they will miss genuine problems?

5. For a given machine family, is one set of spectral alarm bands effectively serving the whole family, or is it necessary to further subdivide the family into two or more groupings? For example, if a utility specified one set of alarm bands for all fans, he will likely find that he needs to further subdivide fans into several individual groups including one set for ID fans, another for FD fans, another for primary air fans, and even another for gas recirculating fans (29).

Finally, it may even be necessary to have one set of alarm bands for the driver and another set for driven pieces of equipment (as was done in Table III, for example, on motor-driven pumps). Or, due to a particular vibration behavior for another family, he may have to specify one set of alarm bands for axial measurements and another for radial (and possibly even different alarm bands altogether for axial, horizontal and vertical measurements).

In general, closely examine each set of spectral alarm bands for each machine family on a quarterly basis (maximum 6 month intervals). Remember, these spectral alarm bands are your **“lifeline”** with the specific purpose of detecting machine anomalies when they occur, and doing so early in the failure process so that corrective measurements can be orderly planned and so that required replacement parts can be acquired.

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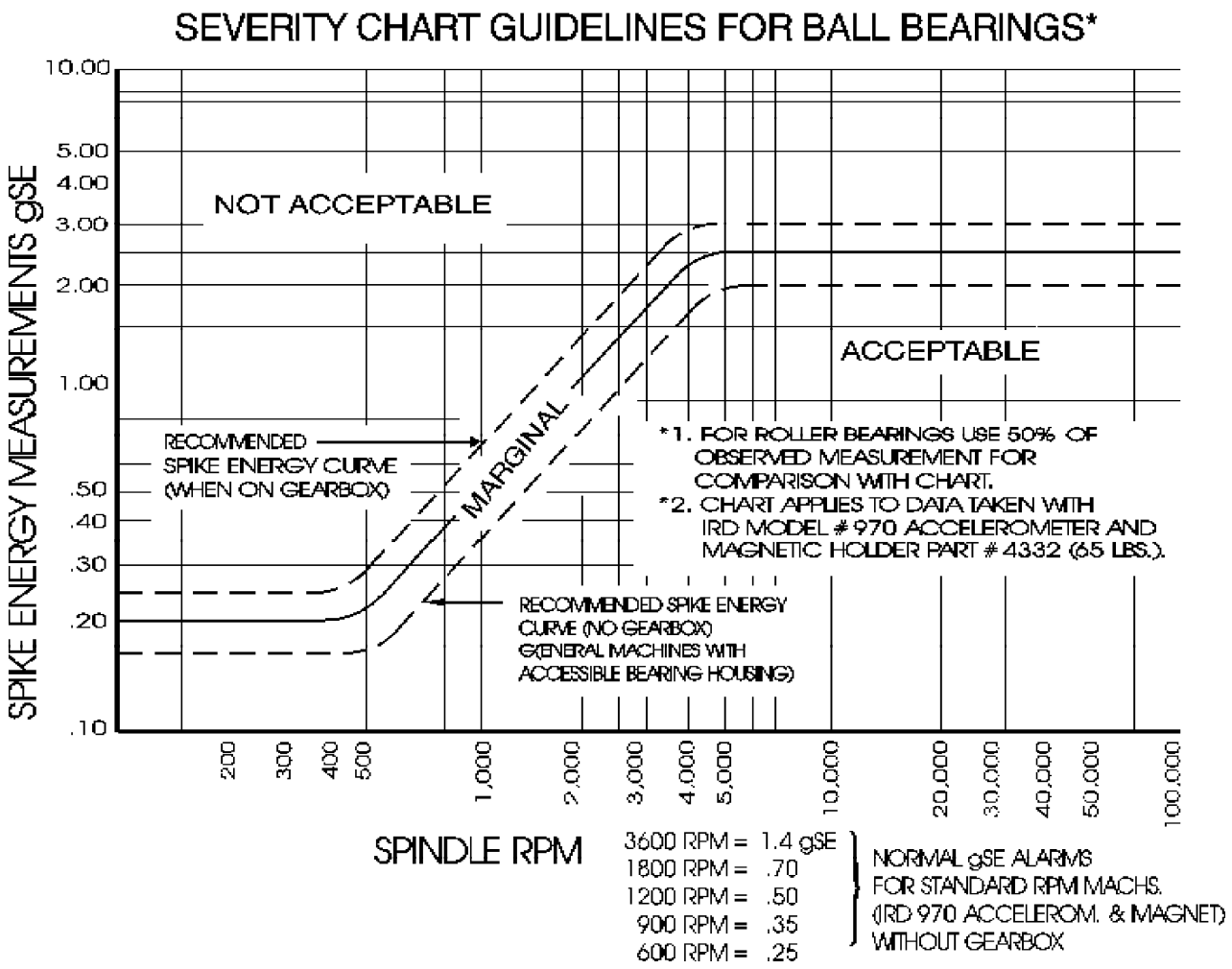
CHART 1
RECOMMENDED SPIKE ENERGY SEVERITY CHART (IRD SPIKE ENERGY)



SPIKE ENERGY™ MEASUREMENTS

Energy is generated by repetitive transient mechanical impacts. Such impacts typically occur as a result of surface flaws in rolling-element bearings or gear teeth.

This energy is conducted from its source through various paths to the outer surface of the machine structure, and is seen as a small-amplitude vibration at the surface. Accelerometers coupled to the surface generate corresponding electrical signal.

The accelerometer signals processed by unique filtering and detection circuitry to produce a single "figure of merit" related to the intensity of the original impacts. This figure of merit is expressed in "gSE" units.

SPIKE ENERGY™ gSE readings are measurements which can with experience, be correlated with the severity of the casual surface flaws. Even though gSE readings are affected by the nature of the conductive path between the impact source and the accelerometer, similar machine structures will provide a reasonable basis for comparison between the structures.

The gSE figure of merit has proven to be effective in detecting mechanical defects in meshing gears and rolling element bearings. The gSE measurement, when used in conjunction with conventional measurement of vibration velocity and acceleration, provides early indications of mechanical deterioration.

** When used with magnetic holders, accelerometers must be installed with a light coating of silicone grease and tightened to 40 in-lb. torque.

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CHART 2
MAINTENANCE DIAGNOSTIC VIBRATION AND FILTERED HIGH FREQUENCY GENERAL TOLERANCE CHART FOR PROCESS MACHINERY WITH ROLLING ELEMENT BEARING①

Unfiltered Vibration Velocity	Shock Pulse & Acoustic Emission	Microlog HFD	IRD Spike Energy②	Severity Level and Alarm Status
Inches/Second Peak	dB	g	gSE	
1.5+	50+	5.0+	3.0+	Danger (Shutdown)
.75 - 1.49	40 - 49	3.0 - 4.99	1.50 - 2.99	Very Rough (Alert)
.40 - .74	30 - 39	1.50 - 2.99	.80 - 1.49	Rough (Alert)
.20 - .39	20 - 29	.75 - 1.49	.40 - .79	Fair (Acceptable)
.10 - .19	10 - 19	.30 - .74	.20 - .39	Good (Acceptable)
.01 - .09	1 - 9	.01 - .29	.01 - .19	Smooth (Acceptable)

① Provide by Charles Berggren of Monsanto
② Spike Energy Amplitudes measured using an IRD 970 accelerometer outfitted with IRD's 2-pole, 65 lb. magnet.

Properly specified, these spectral alarm bands can and will save one many a headache (not to mention considerable dollar savings as well).

7.161 Procedure For Evaluating the Effectiveness of Specified Overall Alarm Levels and Spectral Bands:

Statistical analysis should be employed on both overall alarm levels and spectral band setups for each family of machines. The advent of the personal computer and statistical software now allows one to perform powerful statistical analysis on large numbers of data points that formerly would have been impractical (24, 32, 33).

The procedure for evaluating overall alarm levels is summarized as follows:

1. Take each family of machines and work with the latest data captured on each machine in that family (for example, a utility might work with the data taken on each point of the ID fan family).

2. Calculate the average overall level for the family (X_{ave}) by summing the level for each point on each machine in the entire family and dividing by the total number of samples as per the following formula:

EQUATION (2)

$$X_{ave} = \frac{\sum_{i=1}^{n} X_i}{n} = \frac{X_1 + X_2 + X_3 + ... + X_n}{n}$$

Where:

X_{ave} = Average or Mean Value
X_i = Value of each individual sample i
n = Number of Samples

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3. Calculate the Standard Deviation (S) for the data in this family as per Equation (3):

EQUATION (3)

$$S = \sqrt{\frac{\sum_{i=1}^{n} (X_i - X_{ave})^2}{n-1}} = \sqrt{\frac{(X_1 - X_{AVE})^2 + (X_2 - X_{AVE})^2 + ... + (X_n - X_{AVE})^2}{n-1}}$$

Where:

S = Standard Deviation

4. Then, calculate the Statistical Overall Alarm for the family using the following formula:

Statistical OA = X_{ave} + 2S **EQUATION (4A)**

(some prefer OA = X_{ave} + 3S) **EQUATION (4B)**

Assuming a normal distribution, approximately 95.5% of the data should fall within 2 standard deviations of the statistical average, whereas 99.7% should fall within 3 standard deviations. As the notes accompanying Equations 4A and 4B state, research and application has shown that a statistical overall equal to only the average level (X_{ave}) plus 2 Standard Deviations should be used during Baselining and the first few follow-up surveys before the plant has had a chance to take corrective actions (since vibration levels will be higher at that time. Later, after corrective actions have been taken (which should lower vibration levels), Equation 4B should be used adding 3 Standard Deviations to the mean value for the Statistical Overall.

5. Compare the formerly specified overall for this family of machines to the calculated statistical overall alarm. If significantly different, normally choose an alarm level close to the statistical alarm since it is based on actual measurements on your specific family of machines. However, if it is felt that the current statistical alarm was calculated from an extraordinarily high number of machines either in very good condition, or vice versa, in very bad operating condition, then go with an overall alarm between the specified and statistical, but weighted towards the statistical. (Please see the example case history in Table IV which includes several "aberrations" out of the 25 different machine "families" investigated at 4 different plants).

This same procedure can be used for evaluating the effectiveness of each of the spectral alarm bands, whether using 6 bands, or up to 400 in those systems which allow narrowband alarms. In these cases, take one band at a time; calculate the average for the entire family; then calculate the standard deviation for that band; then calculate the statistical level for that band adding either two or three standard deviations to the mean value as desired. Then, compare the calculated statistical level to the alarm level for that band which had formerly been specified, and adjust as desired. Computer software is available for automating this statistical analysis process.

7.162 EXAMPLE - "Statistical Analysis of Overall Vibration Velocity in 4 Client Power Plants Using the Procedure Recommended Above":

Table IV includes results of a statistical analysis performed on overall velocity levels from 25 different machine types ("families") in each of 4 power plants for a client. Data from 442 machines were analyzed as per the statistical procedure outlined above. Not only were the overall levels themselves analyzed, but also the spectral band setups for each of the 25 machine families as well. However, for brevity, only the analysis of overalls is presented here.

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Referring to Table IV, note that each of 4 items were tabulated for each plant. First, the quantity of machines in each family was recorded. Next, the overall alarm level used by this plant on each machine family was documented. Then, the average overall for each family was calculated. As few as 5 data collection points were on some machines including motor-mounted fans, whereas as many as 40 data collection points were on some turbine/generators. At this point, no effort was made to separately analyze axial and radial readings, but this step is now being taken. Note that the analysis did separately evaluate turbine/generator gearboxes and exciters from the turbine/generators themselves. Finally, note that the standard deviation was computed for each of the average overall levels.

In general, the study showed that overall alarms used by the client during his baselining were within reason. However, adjustments had to be made to a few.

First, 2 families were found to have excessively high vibration which resulted in high statistical alarms. The 8 turbine/generator gearboxes analyzed had an average .423 in/sec overall and a very high standard deviation of .248 in/sec which resulted in a statistical alarm of .919 in/sec. Similarly, the 7 motor-mounted fans had an average .310 in/sec overall and a high .196 in/sec standard deviation (giving a .701 in/sec statistical alarm using 2 standard deviations). The actual spectrum analysis performed on each of these 2 families of machines found very serious problems in a large proportion of them. Therefore, it was felt that the measured amplitudes were unrealistically high. In the case of the 8 turbine/generator gearboxes, the “recommended alarm” (in the far right-hand column of Table IV) was set at .500 in/sec (up from .440 in/sec). Similarly, the motor-mounted alarm was increased slightly from .325 to .375 in/sec.

On the other hand, several families had much lower vibration than the overall alarms specified by the client. This included each of 7 turbine/generators operating at 1800 RPM; 12 turbine/generators operating at 3600 RPM; 45 vertical condensate/circulating water pumps; 32 forced draft fans; 32 induced draft fans; and 9 general purpose gearboxes. For example, in the case of the turbine/generators, the client used the overall alarms now being proposed by the latest version of ISO 2372 now under consideration which it is understood will call for .300 in/sec peak for 1800 RPM machines and .440 in/sec peak for 3600 RPM turbine/generators.

Here, the average levels were only .097 in/sec for 1800 RPM machines and .147 in/sec for the 3600 RPM turbine/generators. Each of these families had fairly low average standard deviations of overalls which resulted in statistical alarms of .226 in/sec for the 1800 RPM and .339 in/sec for the 3600 RPM turbine/ generators (see Table IV). Therefore, recommended alarm levels for both machines were dropped from .300 in/sec to .250 in/sec for the 1800 RPM machines and from .440 in/sec to .350 in/sec on the 3600 RPM machines. These will be further adjusted, if needed, in the future. Similar steps were taken on the vertical condensate/circulating water pumps, forced draft fans, induced draft fans and general purpose gearboxes.

One family obviously had statistically poorly grouped readings with an average overall for 18 general purpose fans being .175 in/sec compared to its standard deviation of .203 in/sec, far above the anticipated norm for a standard deviation compared to this average. Notice that standard deviation levels at both plant 3 and plant 4 were high on these machines. As a result, it was recommended to leave the alarm setting at .375 in/sec until further data has been recorded, and until several machines have been repaired (the spectrum analysis also found a number of problems in these machines which required corrective action).

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Finally, please note the bottom row of Table IV entitled “TOTALS”. Notice that the average overalls for all machines at each plant were all fairly close to one another ranging from a low of .097 in/sec at Plant 2 up to a high of .128 in/sec at Plant 4. Plant 4 had the largest standard deviation of .120 in/sec. Interestingly, after analyzing data from thousands of points on these 442 machines, the final calculated statistical alarm for all of these machines came out to be a level of .294 in/sec - very close to the so-called “industry standard” of .300 in/sec which many plants employ as an alarm on all families of machines, regardless of their machine type, speed or drive configuration. However, by examining Table IV, there still was a great disparity between the average overalls of some families.

7.17 CONCLUSIONS

The last few years have brought about dramatic accomplishments and innovative new, powerful tools into the fields of vibration analysis and predictive maintenance. In addition, the rapid development of the personal computer continues to provide the creative team of vibration analyst and computer programmer with an exciting new menu of imaginative ways of not only storing and processing the vast amount of data being gathered, but of effectively putting it to use. This has the effect of multiplying the accomplishments of those within the vibration diagnostics field, and goes far in expanding the horizons of our knowledge. Surely, many more machine “problem sources” will be added to the diagnostic chart of Table I within just the next 5 years, and some of the information now in the chart will be refined. When this occurs, one will be enabled to set up even more effective spectral alarm bands than those provided in Table III. No matter what happens, the effectiveness of every program will depend on the capability of that system to first detect the presence of real problems, and then all available means including software and the knowledge of vibration analysts themselves will be deployed to diagnose both the cause and severity of the problems. Therefore, it now appears that the success of all future systems will depend on how effective the spectral alarm band setups will be in accomplishing this critical problem detection process. It is sincerely hoped that this paper will be of some assistance in helping many to compose exactly this quality of effective spectral alarm bands.

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TABLE IV
STATISTICAL ANALYSIS OF OVERALL VIBRATION VELOCITY IN 4 CLIENT POWER PLANTS @ 3/28/90

MACHINE TYPE	PLANT 1				PLANT 2				PLANT 3				PLANT 4				CORPORATE SUMMARY (4 PLANTS)					
	QTY.	OA ALARM	OA AVE.	STD. DEV.	QTY.	OA ALARM	OA AVE.	STD. DEV.	QTY.	OA ALARM	OA AVE.	STD. DEV.	QTY.	OA ALARM	OA AVE.	STD. DEV.	QTY.	OA ALARM	CORP. AVE. OA	CORP. AVE. S.D.	STAT. ALARM	RECOM. ALARM
TURBINE GENERATORS																						
A) 1800 RPM	--	--	--	--	2	.300	.078	.041	5	.300	.105	.074	--	--	--	--	7	.300	.097	.065	.226	.250
B) 3600 RPM	4	.440	.161	.093	4	.440	.139	.092	3	.440	.158	.119	1	.440	.084	.055	12	.440	.147	.096	.339	.350
TURB/GEN. GEARBOXES	4	.440	.286	.291	2	.440	.401	.157	2	.440	.718	.253	--	--	--	--	8	.440	.423	.248	.919	.500
TURB/GEN. EXCITERS	4	.300	.153	.082	6	.250	.044	.049	5	.250	.092	.111	--	--	--	--	15	.263	.089	.078	.246	.250
BOILER FEED PUMPS																						
A) MOTOR	4	.300	.113	.067	6	.300	.141	.078	9	.375	.155	.092	--	--	--	--	24	.328	.136	.079	.294	.300
B) TURBINE	--	--	--	--	2	.300	.148	.079	--	--	--	--	2	.300	.233	.127	4	.300	.191	.103	.397	.375
BOOSTER PUMPS	--	--	--	--	2	.300	.087	.032	2	.300	.056	.043	2	.300	.144	.049	6	.300	.096	.041	.178	.200
COND/CIRC WATER PUMPS																						
A) HORIZONTAL	--	--	--	--	--	--	--	--	--	--	--	--	2	.300	.090	.054	2	.300	.090	.054	.198	.200
B) VERTICAL	8	.300	.040	.015	11	.300	.065	.092	18	.300	.081	.071	8	.300	.069	.047	45	.300	.068	.062	.191	.200
HOTWELL PUMPS	7	.350	.140	.083	7	.300	.074	.125	7	.300	.064	.051	--	--	--	--	21	.317	.093	.086	.265	.300
SLAG SLUICE PUMPS	--	--	--	--	--	--	--	--	6	.300	.134	.088	--	--	--	--	6	.300	.134	.088	.310	.300
GENERAL PURPOSE PUMPS	12	.300	.128	.092	10	.300	.081	.054	41	.300	.094	.069	21	.300	.127	.123	84	.300	.106	.084	.274	.300
CENTRIF. AIR COMPRESSORS	1	.300	.125	.053	--	--	--	--	--	--	--	--	2	.250	.102	.070	3	.267	.110	.064	.238	.250
RECIP. AIR COMPRESSORS	5	.425	.149	.054	7	.425	.047	.023	10	.425	.197	.214	4	.425	.125	.075	26	.425	.136	.110	.357	.425
FORCED DRAFT FANS	8	.375	.096	.078	8	.375	.060	.040	12	.375	.063	.054	4	.375	.066	.048	32	.375	.071	.056	.182	.250
INDUCED DRAFT FANS	8	.375	.120	.068	8	.375	.045	.024	12	.375	.066	.046	4	.375	.063	.060	32	.375	.074	.048	.169	.200
MOTOR-MOUNTED FANS	--	--	--	--	--	--	--	--	5	.275	.255	.165	2	.375	.446	.272	7	.325	.310	.196	.701	.375
PRIMARY AIR FANS	--	--	--	--	--	--	--	--	11	.375	.132	.159	--	--	--	--	11	.375	.132	.159	.450	.375
ROOTS BLOWERS	--	--	--	--	--	--	--	--	4	.375	.110	.066	--	--	--	--	4	.375	.110	.066	.242	.300
FLY ASH BLOWERS	--	--	--	--	2	.375	.236	.107	--	--	--	--	--	--	--	--	2	.375	.236	.107	.450	.375
AIR HEATERS	8	.375	.070	.056	--	--	--	--	8	.375	.080	.059	--	--	--	--	16	.375	.075	.058	.190	.250
GENERAL PURPOSE FANS	--	--	--	--	--	--	--	--	6	.375	.212	.128	12	.375	.156	.241	18	.375	.175	.203	.581	.375
PULVERIZERS	--	--	--	--	14	.375	.147	.144	26	.375	.128	.094	--	--	--	--	40	.375	.135	.112	.358	.375
GEN. PURPOSE MACHINES	--	--	--	--	--	--	--	--	8	.300	.066	.032	--	--	--	--	8	.300	.066	.032	.130	.200
GEN. PURPOSE GEARBOXES	--	--	--	--	--	--	--	--	7	.400	.099	.080	2	.300	.126	.066	9	.378	.105	.077	.259	.250
TOTALS	**78**	**N/A**	**.121**	**.080**	**91**	**N/A**	**.097**	**.078**	**207**	**N/A**	**.116**	**.098**	**66**	**N/A**	**.128**	**.120**	**442**	**N/A**	**.115**	**.090**	**.294**	**N/A**

7.2 HOW TO SPECIFY NARROWBAND SPECTRUM ALARMS USING STATISTICAL ALARM AND PERCENT OFFSET METHODS

7.21 INTRODUCTION

Recent years have brought forth many advances in PMP software. Consequently, it has been difficult to remain informed and educated about the use and application of many of these features. This article attempts to inform and advise many predictive maintenance (PMP) software users on the applications for and techniques in the use of one of these many new PMP software features: Narrowband Spectral Alarms. Integral to the use and proper application of these alarms is an understanding of what may be the software's most powerful feature: Calculated Statistical Alarms.

Just as the authors realized several years ago that the Spectral Alarm Band capabilities of most PMP software was being under utilized (or totally unused) in most PMP efforts, so it is today with Narrowband Spectrum Alarms. The reason for the slow adoption of these spectral alarm capabilities is twofold:

1) Ignorance of the benefits that can result from their proper application; and

2) Lack of understanding of the proper method for specifying the alarm limits.

In order to help eliminate the confusion related to the use of spectral bands Technical Associates produced **"Proven Methods for Specifying Spectral Band Alarm Levels and Frequencies Using Today's Predictive Maintenance Software Systems"**. This has helped many PMP software users finally begin to apply the full power of their software and realize the associated benefits. These benefits include earlier warnings of impending failure of items such as gearboxes, motors and rolling element bearings, prevention of "false alarms" on large numbers of machines that do not actually have problems, more effective and timely diagnosis of machine anomalies, and greater confidence that potential downtime from machine failure will be reduced. The goal of any PMP analyst should be to properly identify the machines that are in need of immediate attention to avoid unexpected machine failure and proper prioritization of mechanical and electrical problems that will need attention when the repair manpower is available. Narrowband Spectrum Alarms offer the analyst the opportunity to refine his ability to do this.

In spite of all the advantages provided by the correct use of the Spectral Alarm Band features of PMP software, there still exists a number of areas of concern that are not adequately addressed when using 6 to 12 spectral alarm bands. These areas include any machines that, while in normal operation, produce several prominent frequencies in their spectra, such as Rotary Blowers and Screw Compressors. The vibration spectra from these machines are commonly found to have 4X RPM and several multiples of 4X RPM due to their lobe passing frequencies (without any inherent problem present). Spectral bands alone are adequate to cover maybe 80% to 90% of the general machine population, but are simply not adequate to properly "band" each prominent frequency in the spectrum in machines such as these. Another disadvantage of some spectral alarm band systems is that, in the case of variable speed machines, the alarm bands tend to be somewhat inflexible. Most PMP software sets up the bands with specific upper and lower frequency limits regardless of changes in machine speeds (however, some can be specified in terms of running speed orders, which eliminates this problem). For example, this may result in band alarm violations due to process changes which have brought about a slight change in machine RPM while data is being collected. The change in RPM can shift the frequencies of interest out of the band that was established for it and can result in an alarm violation in an adjacent band with a lower alarm limit.

In response to these types of problems, several PMP software developers have introduced a feature generically known as "Narrowband Spectrum Alarms" (or "Narrowband Envelope Alarms").

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As with Spectral Alarm Bands, Narrowband Spectral Alarms have been greatly under utilized. While it is true some further development by the software suppliers is necessary to provide all the possible advantages of using this technique, this development is not likely to take place until more PMP software users begin to take advantage of the currently available features. Technical Associates has found that a great number of PMP software system purchasers make their acquisition decisions based on the "bells and whistles" of the PMP software they evaluate, and yet these very "bells and whistles" (such as Narrowband Spectrum Alarms) most often remain unused. Hopefully, this paper can spur some interest in further utilizing all the analysis power available to the analyst within their PMP software.

7.22 WHAT NARROWBAND SPECTRUM ALARMS ARE

Since PMP software users interested in using advanced analysis techniques such as Narrowband Spectrum Alarms are usually familiar with the use of Spectral Alarm Bands, they often try to understand the Narrowband Spectrum Alarms in terms of the Spectral Alarm Bands. Although Narrowband Spectrum Alarms have been described as a system that gives the user almost an infinite number of Spectral Alarm Bands, (a statement that can help the user visualize the potential advantages of the Narrowband Spectrum Alarm systems), that description is not very helpful in understanding how the system functions. It is a very natural tendency to try and comprehend something new and unfamiliar in terms of something with which one feels comfortable or at least something in which he may have some prior knowledge and experience. In the case of Narrowband Spectrum Alarms, one may find the subject easier to comprehend if he can separate the two concepts.

As a rule, PMP systems available at this time operate on the basis of a 400 line spectrum. While it is true there are PMP systems currently on the market which allow "route" data to be collected with 800, 1600 and even up to 12,800 lines of resolution, the practical limitations of data collection time, data collector storage capacity and database size normally restrict even the users of these systems to 400 to 800 lines of resolution except for unusual circumstances where the finer resolution is truly a necessity. Getting back to the 400 lines being used most often, this 400 line "standard" results in every vibration spectrum collected being divided into 400 individual components. No matter what frequency has been selected as the maximum frequency for the particular vibration spectrum in question, the spectrum will be divided into 400 equal parts. The influence this has on resolution of the spectrum is obvious. If a maximum frequency (F_{MAX}) of 300,000 CPM is selected, each of the 400 spectrum divisions will equal 750 CPM (300,000/400 = 750 CPM/line). If an F_{MAX} of 12,000 CPM is selected, each of the 400 spectrum divisions will equal 30 CPM (12,000/400 = 30 CPM/line). There are methods for specifying the correct F_{MAX} for many types of machines based on such things as machine speed, type of bearings, and whether or not the machine has a gearbox. These methods for specifying the correct F_{MAX} are covered in detail in Table III along with its accompanying explanations instructing on how to specify the Spectral Alarm Bands for these measurements.

Since each vibration spectrum is essentially divided into 400 individual components, why not set an individual alarm level for each of those components instead of setting alarms for large groups of the components (as with Spectral Alarm Bands)? Actually, this idea has been around for some time. Even before the existence of most PMP software, the U. S. Navy had established a system based on this philosophy for shipboard rotating machinery. Because of the types of analysis equipment in use at the time (tape recorders, real-time analyzers, mainframe computers, etc.) and the logistics of handling all the data, the program was not very successful. The continuing advances of personal computer technology and the advent of the hand-held data collector have made this idea a practical reality. It sounds simple, but the most difficult part is determining a method for setting practical, useful alarm points for each of the 400 lines.

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As is the case with software vendors who offer spectral alarm bands, some of those also offering narrowband alarms give the analyst the option of specifying envelope alarms based either on (1) absolute threshold, or (2) total power. Again, absolute threshold systems require at least one peak within an envelope to either equal or exceed the threshold in order for the envelope to "Alarm". On the other hand, "total power" systems can cause an envelope to alarm without any peak within the envelope equalling or exceeding the specified alarm. The total power within an envelope is based on Equation (5) which follows:

$$Power_E = \frac{\sqrt{\sum_{i=1}^{n} A_i^2}}{\sqrt{N_{BF}}} = \frac{\sqrt{(A_1)^2 + (A_2)^2 + (A_3)^2 + ...+ (A_N)^2}}{\sqrt{N_{BF}}} \quad \textbf{(EQUATION 5)}$$

Where:

$Power_E$ = Total Power within a Narrowband Envelope (in/sec)
n = Number of FFT Lines within the Envelope
A_i = Amplitude of Each of the FFT Line
A_1 = Amplitude of 1st FFT Line within the Envelope
N_{BF} = Noise Bandwidth for Window Chosen
1.5 for Hanning Window

$$\text{Which means } Power_E = .8165 \sqrt{\sum_{i=1}^{n} A_i^2}$$

For example, using power banding, if an envelope were 10 FFT lines wide, the squares of amplitudes in each of the 10 lines would be added together; then the square root of this sum would be taken; then the result would be multiplied by .8165 (assuming Hanning weighting was used).

Therefore, it is important for the analyst to know which type narrowband system he is using ("absolute threshold" or "total power"). Each type has its own advantages and disadvantages. Absolute threshold systems are better suited for those situations when a user has a certain limiting amplitude for a certain frequency (1X RPM, 2X Line, Rotor Bar Pass, Blade Pass, Gear Mesh, etc.). Total Power systems are often more adept at detecting problems which manifest themselves as lower amplitude, broadband vibration (such as late failure stage rolling element bearing problems, cavitation, turbulence, etc.).

If given the option of using either of the narrowband envelope types, he should take this into account when specifying alarm levels. In general, he should specify approximately 40% higher alarm levels when using total envelope power techniques for wideband envelopes.

7.23 SPECIFYING THE NARROWBAND SPECTRUM ALARM LIMITS

7.231 General Discussion

Before the days of common availability of vibration spectrum analysis, PMP analysts began developing overall vibration alarms for machinery in their plants. These alarms were often based on industry standards and analysts' experience in their plants with their machines. Industry standards were often more confusing than helpful and left many analysts wondering how to determine proper overall vibration alarm levels. Technical Associates has published its own table of recommended overall vibration alarm limits for various types of machinery based on a combination of experience and a variety of industry standards (see Table II).

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Setting practical alarm limits for a system of PMP software that divided the spectrum into 6 user-definable bands (Spectral Alarm Bands) was a considerably more complex task than developing alarm limits for overall vibration levels (Table III). Imagine, then, the problems associated with setting 400 individual alarms for each vibration spectrum collected on every piece of machinery in your plant. Stop for a minute and consider the huge amount of data this encompasses. Table V compares the number of alarm values that must be archived and tracked for typical machines (assuming 400-line FFT spectra):

TABLE V
COMPARISON OF TOTAL NO. OF ALARM VALUES FOR COMMON MACHINES
(Assumes Triaxial Readings at each Bearing Housing as a Minimum)

MACHINE DESCRIPTION	OVERALL ALARM ONLY	OVERALL & 6 SPECTRAL ALARM BANDS	NARROWBAND SPECTRUM ALARM
Typical 4 Bearing Pump	12	84	4,800
Chiller with Speed Increaser Gears (8 Bearings w/high and low freq. measurements on gear bearings)	24	252	14,400
Centrifugal Air Compressor 8 Bearing Locations (6 on speed increaser requiring 2 freq. ranges)	24	294	16,800

Table V gives a good idea of how much more data manipulation is necessary when working with Narrowband Spectrum Alarms compared to other alarm methods. Needless to say, software and computers capable of manipulating this large amount of data have to be quite powerful; and most likely, the software will present several options to the user for help in generating the alarm levels. This section is not intended to present all conceivable options for setting up practical narrowband alarms, but does intend to review some of the available alarm options and make helpful suggestions on how this technology might be applied.

7.232 Generating Alarms When Setting Up a New Database

There are two occasions when the analyst is most interested in setting up Narrowband Spectrum Alarms:

1. When the analyst is setting up a new database “from scratch” with no data in the database except the baseline vibration spectra.

2. When he is refining a database that has been in use for some time and has a number of spectra stored for each data collection point.

This section deals with Option 1 (i.e., setting up Narrowband Spectrum Alarms with baseline data only; section 7.24 will cover Option 2).

One of the first tasks the analyst should do before attempting to generate Narrowband Spectrum Alarms from his baseline vibration spectra is to sort through all of the machines in his database and group machines together that are similar into what are known as "machine

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families". By “machine families”, it is meant that machines of the same general design, speed and service should be grouped together.

For example:

“Two boiler feed pumps are in use in a particular plant; both pumps are Gould 4-Stage Centrifugal Pumps with 750 horsepower electric motors. The electric motors both operate at a nominal speed of 3600 RPM, but are from different manufacturers. The pumps are mounted on similar bedplates, etc.”

These pumps could be considered similar for the purposes of generating the Narrowband Spectrum Alarms. If Technical Associates’ guidelines for determining the correct F_{MAX} have been followed (see Spectral Alarm Band Section Table III), machines that are considered similar will have the same F_{MAX} programmed into the database for spectral data collection.

7.2321 Example - Setting Narrowband Spectrum Alarms for a Number of Belt-Driven Fans:

(The following is an example of how the analyst can use the PMP software to help determine if machines are similar enough to be grouped together into a "family" for Narrowband Spectrum Alarm generation purposes.)

A plant had a large number of belt-driven, overhung, centrifugal fans. The electric motors for these fans varied considerably in horsepower from approximately 50 HP up to approximately 250 HP, but all had a nominal operating speed of 1800 RPM. The fans, of course, varied considerably in size, but all had two rolling element bearings and 4 blades. In order to determine if these data points were similar enough for use in generating the alarms, all of the vibration spectra from the motor data collection points were overlaid on top of one another on the computer screen. These overlaid points did not include the special motor electrical spectra recommended by Technical Associates (see cases E and F in Table III of the “Spectral Alarm Band” article). All of the motor vibration spectra and the same F_{MAX}, and the overlay plot is shown in Figure 12. Here, the spectra from each point on each motor are all plotted on top of one another (that is, all outboard and inboard bearing motor spectra). As it clearly indicates, these motor vibration spectra are very similar and these motors’ vibration data can be grouped together very successfully for generation of the fan motor Narrowband Spectrum alarms. On the other hand, the fans could not be used together in a group for generation of Narrowband Spectrum Alarms. Figure 13 shows all the fan bearing spectra overlaid on one another. The problem with this overlay is that it is filled with a multitude of small peaks, each of which represent a running speed (1X RPM) frequency, or a multiple of running speed. This is the result of a relatively wide range of fan running speeds ranging from 990 RPM to approximately 1500 RPM. This created not only variation of the 1X fan RPM peaks, but also the fan blade pass frequency range that varied from 3960 CPM to 6000 CPM. This lack of clear discrete peaks in the overlay spectra indicates that the machines are not similar enough to be used together in generating an alarm for this group of fans. Further refinement of the group is necessary. All of the fan’s running speeds should be reviewed and several smaller groups of fan bearings be made. The RPM of each fan in a group should closely match the RPM of all the other fans in the group. Since F_{MAX} is 60,000 CPM with 400 FFT lines, the resolution of these spectra was 150 CPM. Therefore, preferably, groups should be made such that fan RPM within the group does not vary more than 150 RPM (i.e., 990 RPM, 1055 RPM, 1100 RPM, all within one group since maximum speed variation is 110 RPM). Once the analyst has as many machines grouped together as possible, statistical alarms can be generated for each group.

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7.233 Now for the Statistics:

For the example situation (generating Narrowband Spectrum Alarms for machines having baseline spectral data only), the computer software actually performs a statistical calculation for each of the 400 lines across the spectrum. The baseline data for each spectrum included in the group is used (of course, if one has more than just one survey spectrum, calculations can be performed by the software on all historical spectra). Perhaps the best way to illustrate this is to think of a waterfall (or stack) plot which includes each of the spectra in the group. For this exercise, we will be looking at the waterfall plot on a computer screen and we have a cursor that can move across the screen one line of resolution at a time. This cursor, when set at any particular line of resolution, will allow the analyst to toggle up and down the waterfall and read the amplitude on any of the spectra he desires. In order for the analyst to do his own manual statistical calculations for Narrowband Spectral Alarms, he would set the cursor on the top spectrum at the first line of resolution and record the indicated amplitude; then toggle down to the next spectrum and record the amplitude for the first line of resolution of the second spectrum. This would continue until all the amplitudes for the first line of resolution for each spectrum in the waterfall plot had been recorded. At this time, two statistical variables called the "mean" (or average) and the "standard deviation" can be calculated for the first line of resolution of all the plots in the group. Most people understand what an average value is and can readily calculate the average when they are given a group of numerical values. Very simply it is the sum of the numerical values given divided by the number of numerical values given, or for those who prefer formulae:

(EQUATION 6)

$$X_{ave} = \frac{\sum_{i=1}^{n} X_i}{n} = \frac{X_1 + X_2 + X_3 + ... + X_n}{n}$$

Where:

X_{ave} = Average or Mean Value

X_i = Value of each individual sample i

n = Number of Samples

Calculating the standard deviation (often abbreviated as "σ"), on the other hand is not so straightforward, or universally understood. One way to get an idea of what standard deviation really means is to think of it as the average amount that the numerical values in a group vary from the average value. In other words, if a group of 20 numbers has an average value of 5, and ten of the numbers in the group are equal to 6 while the other ten numbers in the group are equal to 4, it is easy to visualize that the average deviation for this group of twenty numbers is 1. Since each number in the group differs from the average of 5 by 1, the standard deviation for the group would be close to 1.0. Again, for those who prefer to see the formula:

(EQUATION 7)

$$S = \sqrt{\frac{\sum_{i=1}^{n} (X_i - X_{ave})^2}{n-1}} = \sqrt{\frac{(X_1 - X_{AVE})^2 + (X_2 - X_{AVE})^2 + ... + (X_n - X_{AVE})^2}{n-1}}$$

Where:

S = Standard Deviation

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Obviously, to manually calculate these statistics could become quite time consuming, especially when you consider it has to be done 400 times to acquire the statistical values for each line of resolution across the spectrum. Fortunately, computers are excellent at performing large quantities of calculations, and any capable PMP software system that has a Narrowband Spectrum Alarm feature should be able to handle these statistics for the analyst. However, it is always best for the analyst to have some understanding of what values the software is computing.

Following the computer's calculation of the mean and standard deviation for each of the 400 lines of resolution for the selected group of spectra, the analyst has a large selection of options to specify what the alarm values will be. However, before discussing these options, we should first elaborate on these groups we have established. You may question whether all three measurement directions (horizontal, vertical and axial) should be included in the same group. The answer is a definite "maybe". If you have quite a large number of machines in a group and divide your groups further by direction (axial, horizontal and vertical), you might still have groups of about 30 spectra on which you may wish to make this distinction. It will result in a very refined group of alarms, but may also be quite time consuming. Combining directions should be quite acceptable in most cases. Remember, as long as the predominant frequencies are the same, the statistical calculations on the amplitudes themselves will make the basis for the alarm values meaningful.

Now that the computer software has computed the average value at each of the 400 spectral lines of resolution, it is time to determine what the actual alarm value will be. Generally, the choice will amount to using the average spectrum as a base (called the "Alarm Source Spectrum"), and selecting an offset amount to be added to the average, resulting in the alarm value. This offset to be added to the average can be defined either as a percent of the average, or as the average plus any number of standard deviations the analyst may choose. Which offset option and magnitude is selected at this point may depend on the condition of the machines you have selected. Since we are using the scenario that this is a new database here in section 7.23, it may also be true that this is a new plant or a recent expansion area with all new machinery. With all new machines in the group, there are some analysts who feel that the baseline data collected may not necessarily be representative of a true cross section of machine conditions (i.e., some machines in average health, some machines in poor health and some machines in excellent or new condition). This assumption would then lead one to think that the standard deviation calculation may not be as meaningful for this application as in an older plant. They would recommend choosing a number such as 50% offset above average as the more applicable alarm for these cases. Technical Associates' experience has been that even when all equipment is new, standard industrial machines that are found throughout most plants commonly have a significant number of mechanical and electrical problems. This means that in almost any conceivable situation, the average value plus a number of standard deviations should be a good choice for the alarm value (however, see further comments regarding a minimum threshold below).

Before a decision is made on how many standard deviations that the alarm value should be offset from the average, it would be wise to learn a little more about the field of statistics and vibration analysis. Statistical analysis of overall vibration levels has been performed manually for many years to refine overall vibration alarm values. The result has been that the average value plus two standard deviations (or 2σ) normally produces what can be considered a very effective overall alarm value. This 2σ value above average has also been successfully applied to Spectral Alarm Band power values (when the Spectral Alarm Band software calculates power in the band rather than the "peak" value or threshold alarm). It should also be noted that some analysts have encountered circumstances where they preferred to set the alarm values at the average plus 3σ for overall vibration alarm values. The question remains: "How much above average should the Narrowband Spectrum Alarm be adjusted?" A study of statistics reveals that for a normal distribution, 95.5% of the data should fall within 2σ of the

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average (within the envelope of ±2σ from average), and 99.7% should fall within 3σ of the average (again, within the envelope of ±3σ from average). As vibration analysts we are only interested in the data that exceeds the alarm value. In a normal distribution this should be half of the numbers quoted above (since half would be above average and the other half below average). So with overall vibration alarms, a typical 4 bearing fan having 12 measurement points and an alarm value set at 2σ above average would yield an alarm violation for only 2 measurements out of every 8 machines tested.

That is, if each machine has 4 bearings; and each bearing has triaxial vibration readings, this equals 12 measurements per machine. 8 machines times 12 measurements equals 96 total points. 4.5% of data falls outside ±2σ, so only 2.25% will exceed 2σ above average. 96 measurements times 2.25% equals 2.16 measurements greater than 2σ above average.

Does this mean 2σ above average will work just as well for Narrowband Spectrum Alarms? Remember from Table V that a typical 4 bearing machine will have 4,800 individual alarm points assuming 400 FFT lines. Multiplying 4800 by 2.25% will give us the portion of these 4800 alarm points that will likely be violated (4800 X.0225 = 108). Does this mean that 108 alarm violations will occur for even the simplest machines? Well, perhaps the 2σ above average alarm value is not ideal for Narrowband Spectrum Alarms. Certainly we can't deal with every machine being in alarm. As we said at the beginning, the PMP vibration analyst wants to correctly identify real machine problems - not spend his time sorting out false alarms.

In reality there are reasons that 108 Narrowband Spectrum Alarms will not occur on every simple 4-bearing machine. The first reason is that the Narrowband Spectrum Alarm software offers a function called "enveloping". When the envelope function is invoked, the width of the alarm envelope around the spectral peaks is altered as specified by the analyst. The analyst has the choice of "Constant Bandwidth" and "Constant Percentage Bandwidth" envelopes. "Constant Bandwidth" is selected for machinery items such as electric motors and machines direct-coupled to electric motors where RPM is very constant from survey to survey. Technical Associates generally specifies a bandwidth of 10 lines of resolution around 1X RPM and other peaks in these cases. This means that in both the high frequency and low frequency portion of the vibration spectrum, the envelope will be 10 spectral lines of resolution wide. (Note: For some machines such as Turbine/Generators that have very severe constant speed requirements where relatively slight RPM changes have adverse impacts on product quality, the analyst can select a narrower envelope, such as 5 or 6 spectral lines, so that when undesirable RPM changes occur, the spectral peaks will fall outside their alarm amplitude and thereby trigger an alarm. This may help troubleshoot quality and process problems).

Figure 14 shows a statistically calculated (average plus 3σ) Narrowband Spectrum Alarm developed from all measurement point baseline spectra on two identical reciprocating direct-coupled air compressors with the envelope function disabled. The spectral data, though not unusually high, is difficult to distinguish from the alarm itself. If the analyst attempted to use an alarm like this, just about every machine spectrum would be in alarm. Figure 15 shows the exact alarm (average plus 3σ), but with a Constant Bandwidth of 10 spectral lines selected. The difference makes the alarm usable. A comparison of the two alarms in Figures 14 and 15 make it easy to see how use of the envelope function will greatly reduce the number of alarm violations from what statistical calculations alone indicate they would be. As shown earlier for an average plus 2σ alarm, normal data distribution would indicate 108 spectral violations for a typical 4 bearing machine without the enveloping function invoked. When an average +3σ is used, .15% of the data could be expected to exceed the alarm. Multiplying 4800 times .0015 yields 7.2. Thus, this shows that without enveloping function used, every simple 4 bearing machine tested would have at least 7 Narrowband Spectrum Alarms violated with an average +3σ offset.

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"Constant Percentage Bandwidth" is normally invoked for machines that commonly have some RPM variance. This may include belt-driven machines where belt wear and adjustment normally allow some RPM changes, or even variable speed machines that make small speed adjustments for process variables. Since relatively minor speed changes can produce large changes in frequency at higher multiples of RPM, even a Constant Bandwidth envelope may be violated in the high frequency part of the spectrum when an RPM change occurs. Though the direct-coupled reciprocating compressor used in Figures 14 and 15 would not normally have a Constant Percentage Bandwidth envelope invoked, Figure 16 has been included to illustrate the effect of the Constant Percentage Bandwidth. It shows the same average $+3\sigma$ alarm value, but an obvious change has occurred. With Constant Percentage Bandwidth invoked, the envelope gets continually wider in direct proportion with increasing frequency. Thus, a small change in machine RPM, which may be expected on some machines, will not result in a Narrowband Spectrum Alarm envelope violation by a higher order multiple of running speed. This is very useful on variable speed machines because a speed change of 100 RPM on a 1800 RPM machine means the fundamental (1X RPM) frequency changes only 100 CPM, but the 10X RPM multiple of running speed will have a corresponding 1000 CPM shift in frequency.

So far it has been demonstrated that the number of alarms violated when using the statistical features of Narrowband Spectrum Alarm software can be reduced through using the envelope feature of the software. This is effective whether using Constant Bandwidth envelopes, or Constant Percentage Bandwidth envelopes. Unfortunately, the analyst will find that even with the envelope function invoked and properly adjusted, far too many alarm violations will occur at average $+2\sigma$, at average $+3\sigma$, at average $+4\sigma$ and even at average $+5\sigma$. At this point (average $+5\sigma$), the Narrowband Spectrum Alarm envelope is usually approaching excessive amplitudes such as a 1.0 inch/sec velocity level around the 1X RPM fundamental frequency, which in most cases would be considered an unacceptably high alarm value. Yet, the analyst will still see a very high number of Narrowband Spectrum Alarms being violated.

A look at which portion of the spectrum is producing these alarm violations may be in order. As you study typical vibration spectra, a pattern emerges. Of course this pattern is the basis of all the vibration diagnostics work done. The pattern generally consists of a relatively large discrete peak of vibration energy at 1X the machine running speed, usually a smaller peak at 2X running speed and then, depending on the machine type, there may or may not be a few more discrete frequencies in a healthy machine's vibration spectrum. Beyond these discrete frequency peaks, the spectral energy goes to virtually zero with only small amounts of random vibration (when the machine is healthy). This is very evident in the higher frequency portions of most vibration velocity spectra (see Figure 15, particularly between about 31,000 and 45,000 CPM and 50,000 to 60,000 CPM with a near zero amplitude alarm). With no consistent energy in these areas of the spectrum, the statistical processing performed by the software essentially averages these areas of the spectrum to a value of zero. This means that the statistically produced alarm values for parts of the vibration may simply be the result of using a basis of the average (zero) and adding several standard deviations (also equal to little more than zero), and coming up with an alarm level equal to little more than zero. Of course, every time vibration data are collected, some spurious data will be included. Whether it is from the instrumentation itself or from a passing railroad train, it is obvious that the statistically calculated alarm value of near zero will result in numerous alarm violations.

Therefore, Narrowband Spectrum Alarm software offers another feature to eliminate this problem. It is called the "user specified minimum alarm value" (or "threshold"). At this point the discussion is dealing with setting up the Narrowband Alarm values using only baseline data. Therefore, the analyst may be able to determine an adequate minimum alarm value by carefully reviewing the overlaid spectra shown in Figure 12. From visually studying this plot,

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one might quickly conclude that a good minimum alarm value would be approximately .035 in/sec, but using this method may require adjusting the minimum alarm level several times to achieve the desired results. For example, Figure 17 is the same as Figure 15 with the addition of a user-defined minimum alarm value equal to .015 in/sec. Notice how some of the lower level envelopes centered at about 22,000 and 46,000 CPM have disappeared because the minimum threshold is now set higher than the calculated 3σ alarm would be.

As always, the proper minimum specified Narrowband Spectrum Alarm values are highly dependent on the type of machinery being tested. While a minimum specified alarm value of .03 in/sec may be entirely appropriate for large motors on overhung belt-driven fans, it would be much too high for high-precision machine tools. Figure 19 illustrates how General Motors has adjusted various spectral alarm band levels in accordance with the type of machine being tested (Figure 19 assumes power banding).

Up to this point, stating the preferred offset for the alarm (above average) has been avoided. Technical Associates has found that when the proper minimum Narrowband Spectrum Alarm value is specified, a Narrowband Spectrum Alarm value of the average $+3\sigma$ works well. It must be pointed out that these are recommended beginning points for your Narrowband Spectrum Alarm efforts. Once you gain experience and confidence (in addition to more spectral data), you will definitely want to refine your alarms further to more adequately serve your needs.

7.234 What About Unique Machines that Cannot be Comfortably Grouped Together? (% Offset Alarm Above Baseline Spectra):

The above discussion handles specification of the Narrowband Spectrum Alarms for similar machines in a new database having only baseline spectral data for the most part. Chances are, there are most always a good number of machines that cannot, for one reason or another, be grouped together for common alarm generation (of the whole group). Options for dealing with these machines are somewhat more limited. Since baseline data is all that has been acquired thus far (in this section 7.23), the only reasonable option for generating some type of Narrowband Spectrum Alarm for these machines is to use the baseline spectrum for each point as the basis for the alarm. Rather than specifying the alarm as a certain number of standard deviations above average, the offset can be specified as a specific percentage above the baseline spectrum. As with the alarm specified using statistical methods, the envelope and user-defined minimum alarm features are highly recommended. Make your decisions on the use of these features in the same manner discussed above when using statistical alarm methods. The amount of offset to be used for this type alarm will be something of an educated guess. The value of this alarm is based on the general agreement that if all else remains the same (machine operating parameters, etc.), and vibration levels increase, machine health is deteriorating. An offset from the baseline spectrum of anything less than 50% is probably insufficient to cause great concern since vibration readings from survey to survey quite commonly vary more than 20% just due to common instrumentation and measurement error. Figure 18 is an example (again using the reciprocating air compressor) showing a Narrowband Spectrum Alarm generated using the baseline % offset (sometimes referred to as an “individual” alarm). This alarm method generates a different alarm for each measurement point spectrum. This alarm is valid only for this particular air compressor and only at position 2 Vertical.

If none of the Narrowband Spectral Alarm development techniques discussed thus far are able to generate what the analyst considers an adequate alarm, another option remains. Using the edit alarm function of the PMP software, the analyst may modify the Narrowband Spectrum Alarm by drawing the preferred alarm criteria on the computer screen using the keyboard cursor arrow keys, much in the same way an “Etch-A-Sketch” toy operates. Using this method, total control over Narrowband Spectrum Alarm values is available to the analyst.

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7.24 GENERATING ALARM VALUES FOR A PRE-EXISTING DATABASE

Section 7.232 covered specifying Narrowband Spectrum Alarms for those machines where only baseline spectra are available. This would typically be the case if a new PMP vibration analysis program is being established. It may also apply where the PMP vibration analysis program is being extended to cover more machinery items or recent plant expansion areas. Section 7.232 covered many of the "basics" of the use and generation of Narrowband Spectrum Alarms. This section 7.24 applies when Narrowband Spectrum Alarms are generated for a long-existing database containing a significant amount of spectral data for the machines included in the database. Several options exist for the analyst's use in developing accurate Narrowband Spectral Alarms with a spectral history available that are not usable when only baseline data is available. Also, since the analyst has been dealing with these machines for an extended period, he may have greater confidence in making decisions about which machines can be grouped together for the statistical calculations.

As the above statement indicates, the analyst should, once again, develop machinery groups following the same guidelines established in Section 7.232. It is Technical Associates' opinion that the statistical method is the preferred way to generate narrowband alarms in most cases. When a good spectral history is available, the analyst has a choice of whether to develop the statistical Narrowband Spectrum Alarm unique to each measurement point, or combined with other similar machines. Good arguments can be made for both methods and each method has its advantages. Since comparison with other machines in good health is a long standing and valid method used in machinery diagnostics, Technical Associates recommends grouping similar machines to produce Narrowband Spectrum Alarms common to all applicable measurements on these machines. When this method is used, not only is the most recent spectrum from each measurement point in the group included in the statistical calculation (as in the case of Section 7.232 where only baseline spectra are available), but also all of the spectral data archived for each measurement point will be included in the group. As with any statistical calculation, the larger the population used for the calculation, the more accurate the statistical calculations will be. It is in this situation, then, that the analyst can use the power of the software and computer to achieve the most precision in computing effective Narrowband Spectrum Alarms. Some PMP software programs call this alarm type "Statistical of List".

All of the recommendations concerning the various envelope techniques discussed in Section 7.232 are applicable to this alarm generation method as well. The same technique discussed in Section 7.232 for specifying a minimum Narrowband Spectrum Alarm value also applies. In fact, when applied to this situation, even more precise minimum alarm levels can be specified if the Band 6 Spectral Alarm level has been statistically adjusted since its original setup. The method for statistically adjusting the Spectral Alarm Bands is thoroughly covered in the Spectral Alarm Band article (Table III and accompanying text).

Some PMP analysts who work with massive machines having a large number of rollers and bearings all running at different speeds (i.e., paper machines) may prefer to use another method for establishing their Narrowband Spectrum Alarms. This method is similar to the "Individual" alarm discussed in Section 7.232 in that it generates an alarm that is applicable only to its own unique measurement point. The difference is that it is a statistical calculated alarm (sometimes called "Statistical of Point"). The data used for calculating the alarm values is simply the archived spectral data for each point (visible by looking at a "waterfall" or "map" plot of the measurement point). The resultant alarm is strictly a result of that measurement point's history and lacks comparison to any other data. The alarm applies only to the point specified just as that discussed in Section 7.232 and referencing Figure 18.

The average, plus 3σ alarm value explained in Section 7.232 should be a valid alarm value for both of the aforementioned statistical alarm methods. Of course, the success of this as a valid alarm

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level depends, as it did in Section 7.232, on the proper selection of the user-specified minimum alarm level. The analyst must be constantly mindful of the machine type being monitored and specify a minimum Narrowband Spectrum Alarm level consistent with the machine type being analyzed.

7.241 Specification of Narrowband Spectrum Alarms for Variable-Speed Machinery:

Another type of alarm specification may be valuable in generating Narrowband Spectrum Alarms for variable speed equipment. It is known as a "peak of point" option and, as with the "statistical of point" and "individual" type alarms, this alarm is generated on a "per point" basis, each alarm being valid only for that single point. As with the previously discussed "statistical of point" alarm, the entire spectral history (visible in a "waterfall", or "map" plot) is used to generate the alarm. Rather than performing statistical calculations on the data, the software generates a "peak hold" alarm source spectra. In other words, the maximum amplitude at each line of resolution throughout that measurement point's spectral history is stored and this resultant spectrum is captured and used as the alarm source spectrum for this point (of course, if it is known that any of these machines being evaluated have a noticeable problem, these should be excluded from the "peak hold" spectra exercise). The alarm source spectrum can then be altered using the envelope methods and by specifying a minimum alarm amplitude just as was done with previous alarm methods. The reason that this is very effective with variable speed machines is that the resultant "peak hold" spectrum generated from historically frequency-shifting spectral data very accurately establishes for the analyst exactly in what frequency ranges each discrete frequency peak will operate during normal operation (assuming enough spectral data history is present to clearly define the operating range for the machine). The analyst must again decide which type of envelope to select (most likely "Constant Percentage Bandwidth"since a variable speed machine is being discussed). A look at the vibration source spectrum with the envelope function disabled should help the analyst decide fairly quickly what <u>percent</u> bandwidth will be required to adequately envelope this machine's spectral data points. The analyst must also specify the offset to be added above the alarm source spectrum to establish the alarm value. Since the alarm source spectrum contains the maximum amplitudes from all the spectra at each point, it is advisable to start with a relatively small offset, such as 20% and make adjustments, as necessary.

At this point an important subject should be brought up for discussion. Some of the Narrowband Spectrum Alarm software available does not allow the analyst to determine how much of the spectral history should be used for statistically calculating the alarm source spectrum. Its only option is to use the entire history. This becomes a problem if at some point in the history of the database, some of the refinements made involved a change in the F_{MAX} of the spectra collected (or if the machine has been overhauled resulting in dramatically improved vibration behavior). When the software attempts to make the necessary statistical calculations to generate the alarm source spectrum, it detects that at some point in the spectral history the F_{MAX} was altered and consequently "kicks the software out" of the alarm generation process, leaving a message on the screen that the calculations were not possible. This serious deficiency could be remedied by simply allowing the analyst to specify how many previous spectra to include in the calculations. Hopefully, if enough people begin to use the Narrowband Spectrum Alarm features of their software, inconvenient problems like this will be addressed.

In addition, if a machine has been overhauled resulting in noticeably improved vibration behavior, the analyst should have the option of either looking at the older set of spectra taken when the machine had the problem, or limiting statistical alarm calculations only to those spectra taken after the machine has been overhauled. This is a very important feature that deserves close attention and action on the part of the software vendors.

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7.25 SUMMARY

Though not yet a widely used or understood technology, the power of Narrowband Spectrum Alarms is here for those of us in the Condition Monitoring field to use. As with Spectral Alarm Bands a few years ago, the software developers have provided the power and the technology and have left it up to the vibration analyst to determine the best methods and techniques for using it. This paper has been a compilation of Technical Associates' experience and research on the subject to date. It is intended as an aid in helping the novice Narrowband Spectrum Alarm user have some idea on how to begin effectively using this powerful technology. Certainly, Technical Associates understands the more we use and become comfortable with the Narrowband Spectrum Alarms, the more we will find it may be necessary to revise the recommendations contained herein. Hopefully the information provided will at the very least increase the analyst's understanding of how Narrowband Spectrum Alarms work, where Narrowband Spectrum Alarms can be applied and help remove some of the apprehension analysts feel about invoking this important capability.

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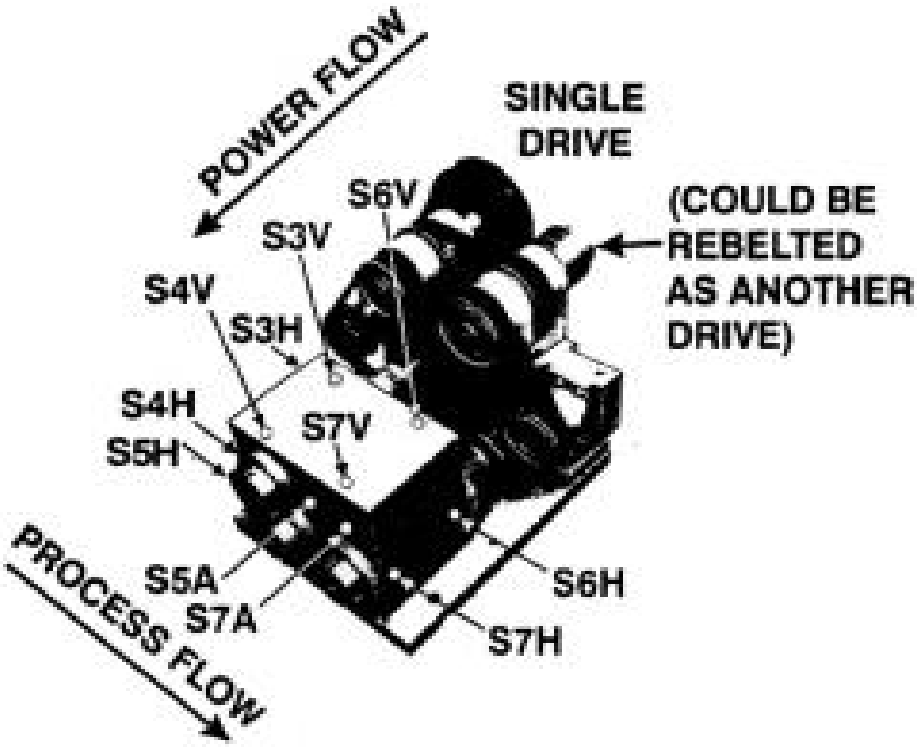

FIGURE 12

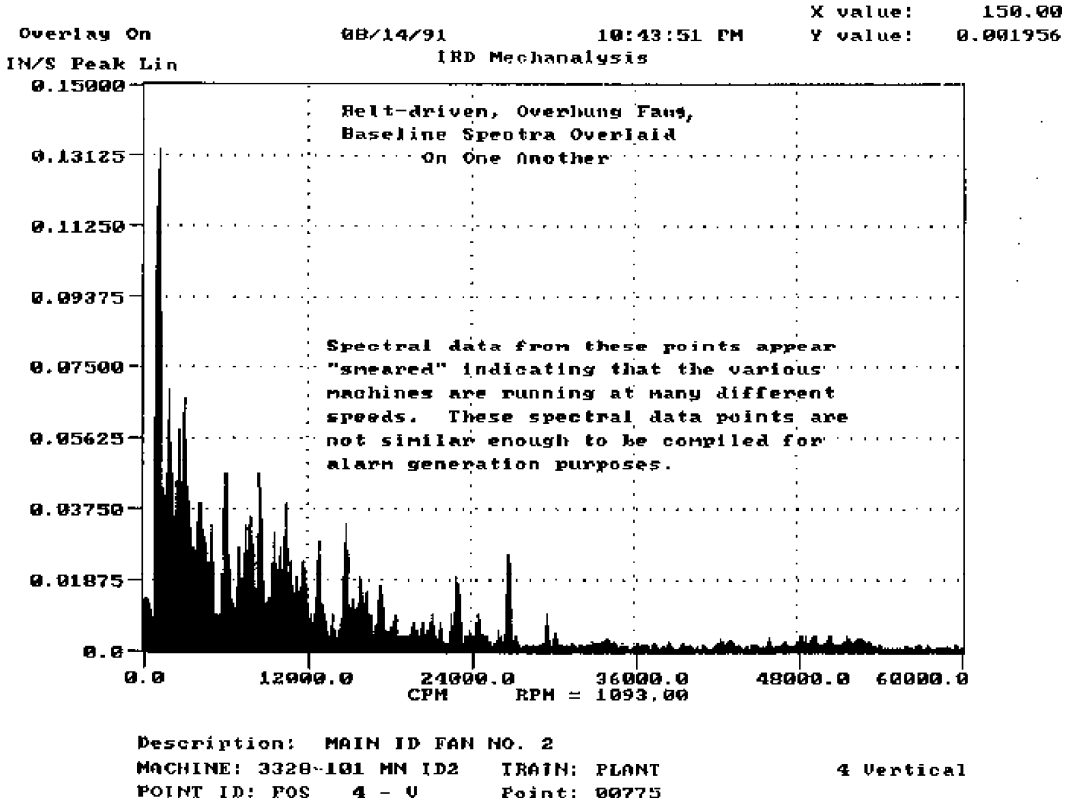

FIGURE 13

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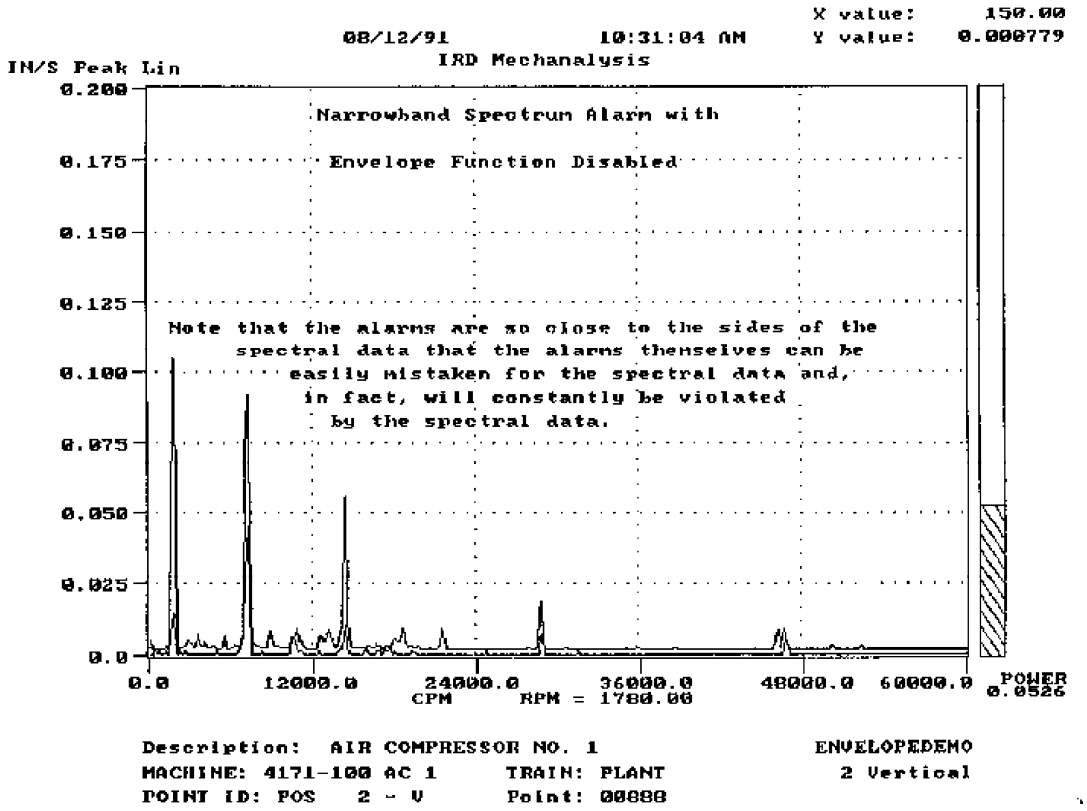

FIGURE 14

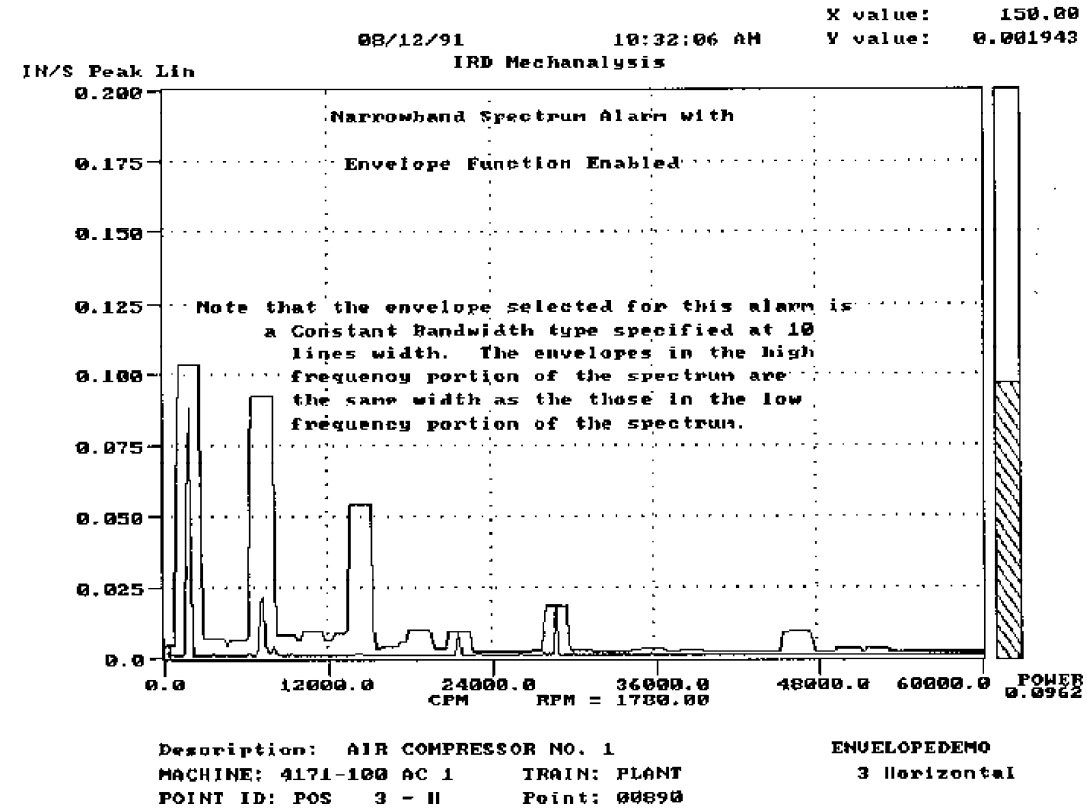

FIGURE 15

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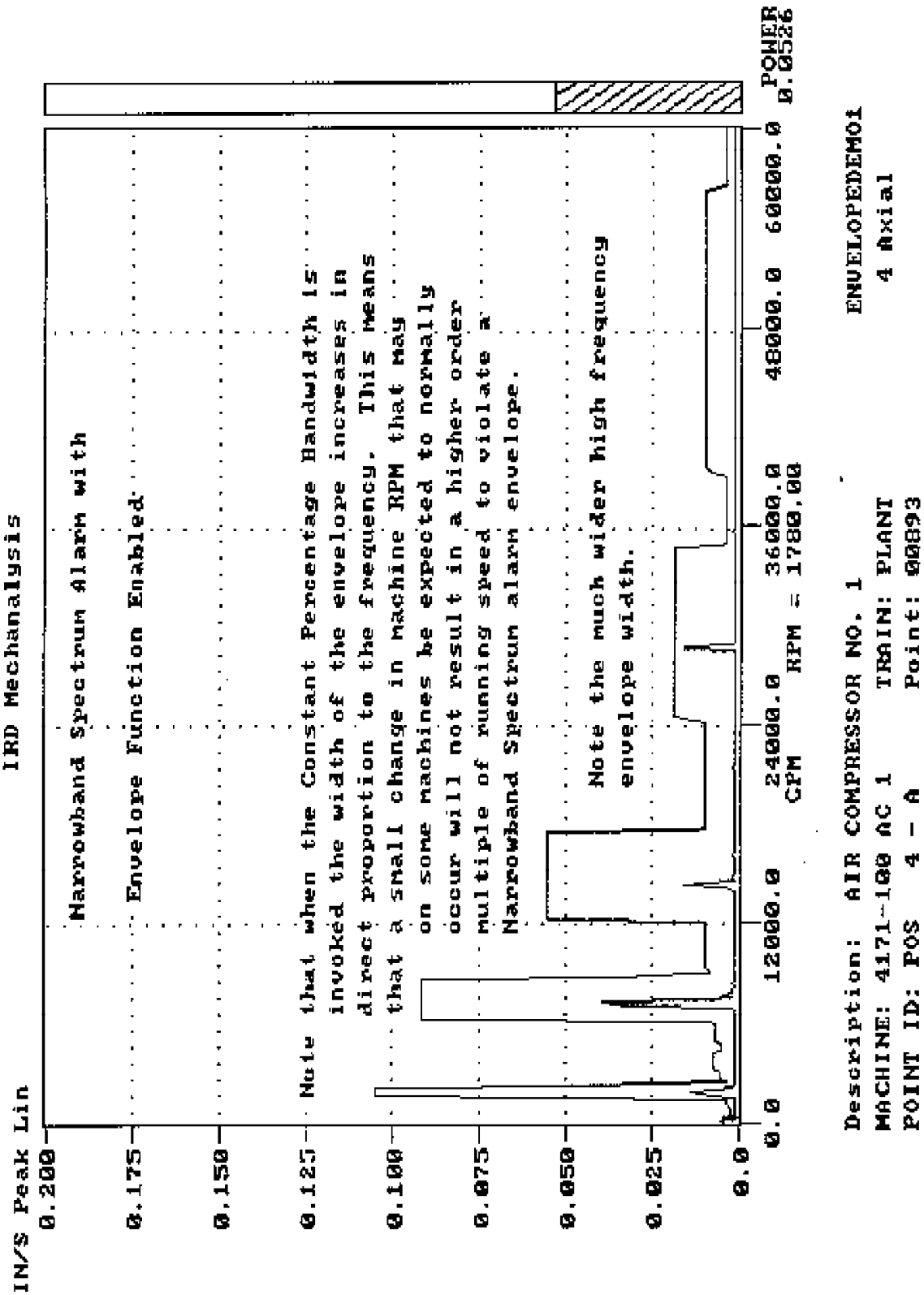

FIGURE 16

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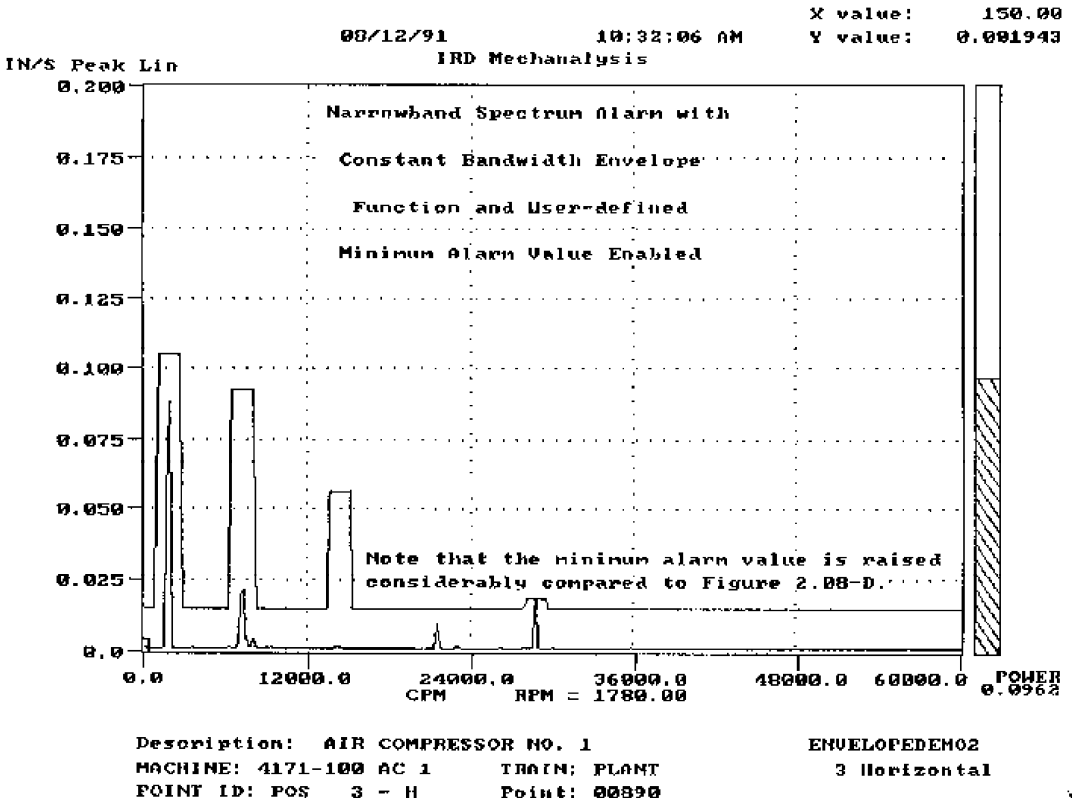

FIGURE 17

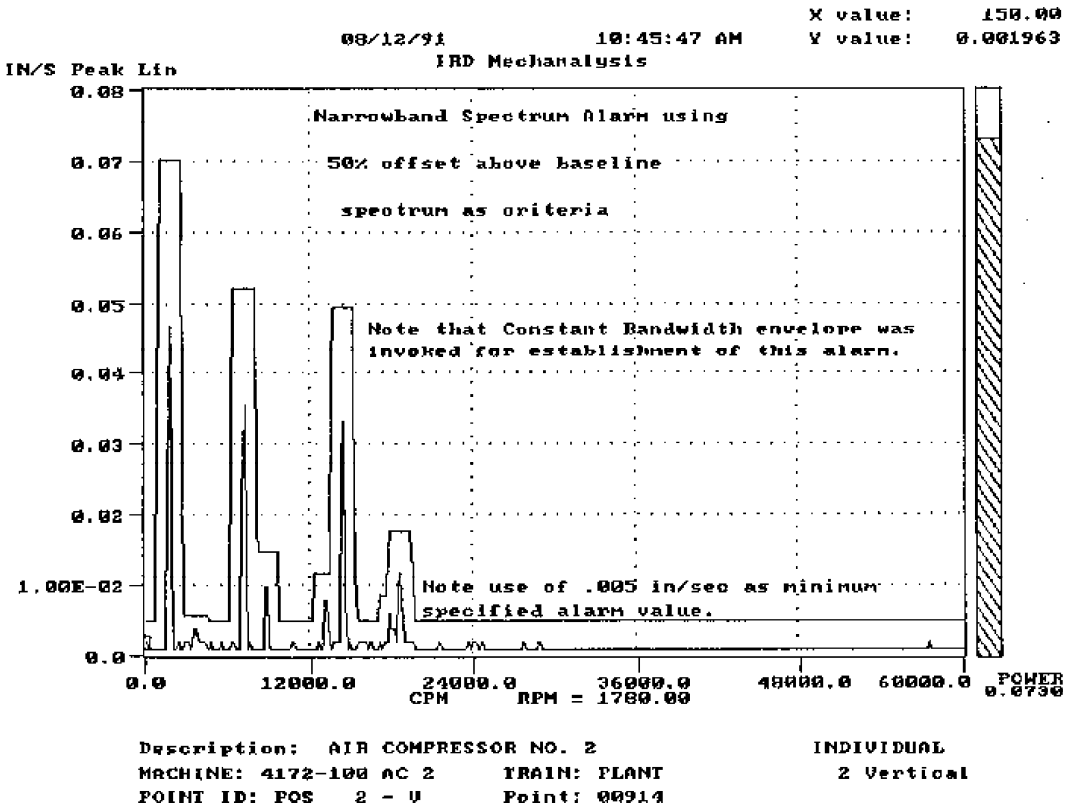

FIGURE 18

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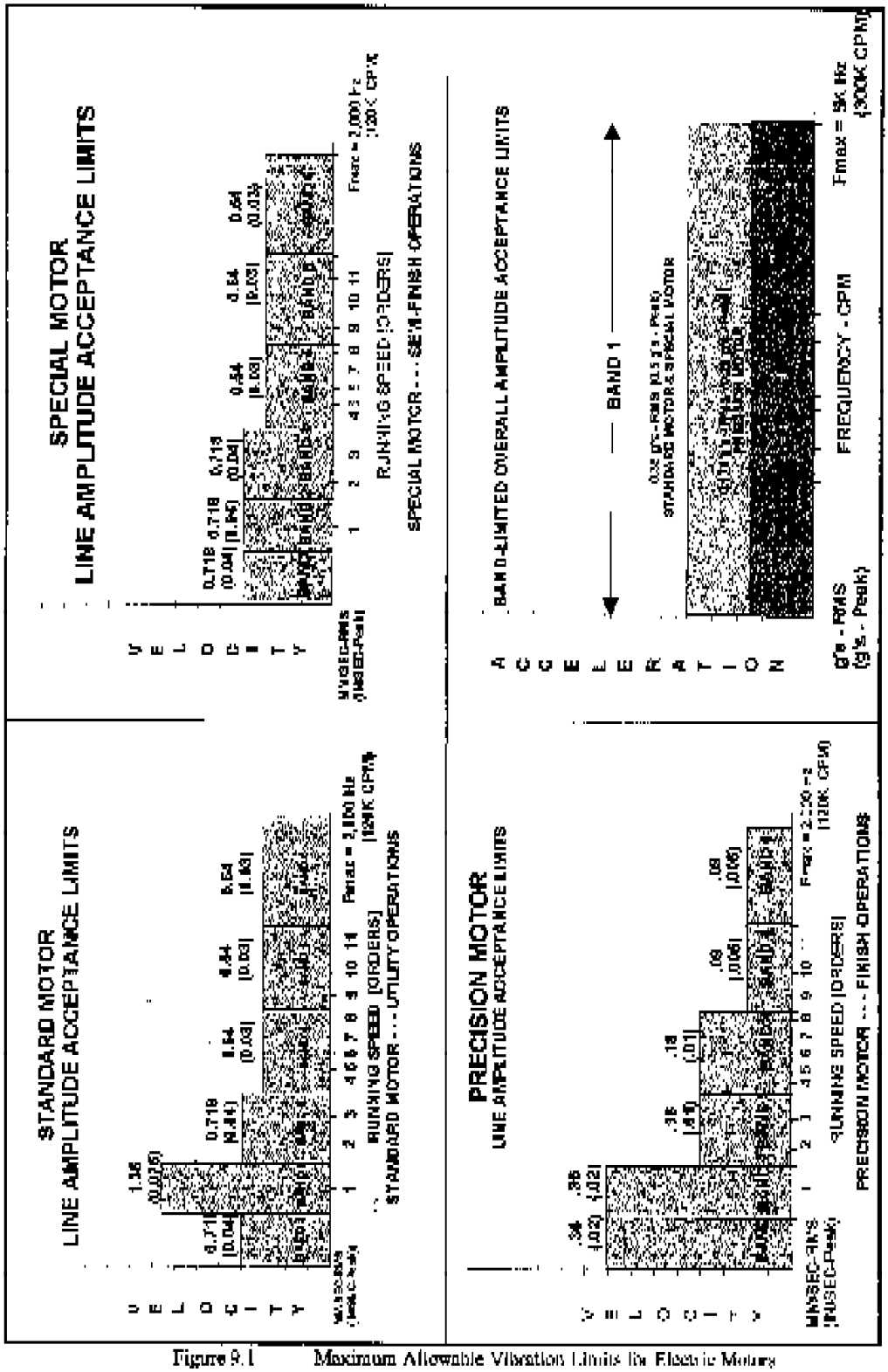

Figure 9.1 Maximum Allowable Vibration Limits for Electric Motors

FIGURE 19
GENERAL MOTORS ACCEPTANCE VIBRATION STANDARDS FOR NEW OR REBUILT STANDARD, SPECIAL & PRECISION MOTORS (@ 1997)

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CHAPTER 8

INTRODUCTION TO LISSAJOUS ORBIT ACQUISITION AND INTERPRETATION

8.0 INTRODUCTION

In recent years, it has become common practice to install dual non-contacting eddy current probes on critical high speed turbomachinery to avoid false shutdowns in the event that one of the probes should fail. The pickups are mounted radially in each bearing with the axis of the pickups separated by 90°. Some are mounted in the true horizontal and vertical directions; others are mounted on an axis 45° from the horizontal at the "10:30" and "1:30" clock positions. Since these non-contacting pickups are already positioned in the bearings, they can be hooked into an oscilloscope to provide Lissajous orbit patterns of the shaft's motion in the bearing.

This chapter will deal with the proper setup of transducers and instrumentation required to produce Lissajous orbits. Some example Lissajous plots will be given and their use in the diagnosis of common machine problems will be provided. The interpretation of Lissajous plots will be the highlight of this chapter.

Since this is an introduction to Lissajous orbit generation and interpretation, advanced techniques and complex situations will not be featured. However it will provide a solid basis for future discussion.

8.01 WHAT IS A LISSAJOUS ORBIT?

A Lissajous orbit is a plot of the shaft's orbit within the bearing corresponding to the shaft's total motion. *This orbit will likewise display the total shaft displacement (mils or microns)*. Perhaps, it is easiest to understand if one focuses on the center of the end of a shaft. If the motion of that center could be magnified and the rotational speed slowed down, one would be able to see how the shaft center is actually moving. Is it jumping erratically? Is it rotating in circular loops? Is it rotating in an elliptical fashion? What does it mean? *The purpose of a Lissajous orbit is to provide a trace of the actual movement of the shaft within the confines of the clearance in a plain bearing.*

Lissajous "plots can be used to detect abnormal movements of the shaft in a bearing. Abnormal movements will often be a characteristic of a developing fault".[1]

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8.02 A TYPICAL SETUP FOR GENERATING LISSAJOUS ORBITS

Figure 1 is a typical setup for generating a Lissajous plot. It features two non-contacting pickups, a signal conditioner, and an oscilloscope (many of today's multi-channel data collectors and spectrum analyzers can likewise generate Lissajous orbits). An alternative setup is shown in the upper portion of Figure 2.

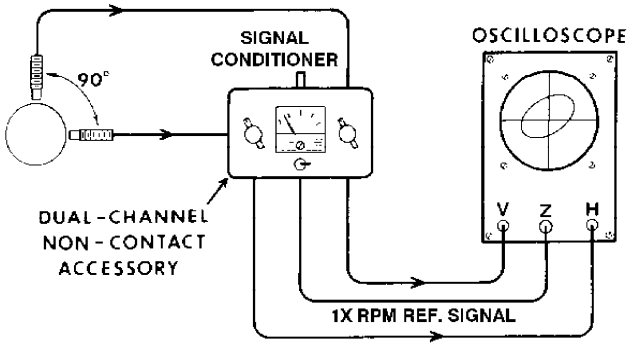

FIGURE 1
A TYPICAL SETUP FOR DISPLAYING LISSAJOUS ORBITS USING NON-CONTACTING EDDY CURRENT PROBES (Ref. 2)

8.03 SETTING UP THE NON-CONTACT PICKUPS FOR LISSAJOUS ORBITS

It would be ideal if the non-contacting pickups were installed with their axis in the true horizontal and vertical directions at the shaft. In many cases however, they are placed 45° from the horizontal to allow for the split lines in the bearing housings. This will not affect the shape of the observed pattern, but it will cause the display to be rotated 45°. However, compensation for this can be made on the CRT screen on the oscilloscope. How this is accomplished will be mentioned later.

It is important that the sensitivities of the non-contacting transducers be known. It is necessary to check the specifications to be sure. The sensitivity rating is given in millivolts per mil. For example, most non-contact pickups have a sensitivity of 200 millivolts per mil. These input sensitivities are matched by the adjusting the gain sensitivities on the oscilloscope. (Ref. 2)

The reason for this information is to ensure that the shape of the Lissajous orbit accurately reflects the true motion of the shaft. It is important that both the horizontal and vertical scales are set up with identical displacement scaling (.1 mil/division; 1.0 mil/division, etc.). Otherwise, all resultant patterns (orbits) will be distorted. *In addition, it is very desirable to ensure accurate amplitude readings from the scope (or analyzer)*.

8.04 PROVIDING A ONCE-PER-REVOLUTION REFERENCE PULSE

A choice is usually made between installing another non-contacting pickup or an electromagnetic pickup close to the shaft to provide a once per revolution trigger pulse. Usually a key or keyway is used to trigger the reference pickup. If there is no key or keyway, a shallow slot can be made in the shaft in a discrete location to trigger the pickup. Other alternatives are available.

If the oscilloscope is equipped with a "Z" axis (intensity) input, the lead from the reference pickup can be inserted there. This will produce one or more "blank" spots on the Lissajous orbits.

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8.05 THE OSCILLOSCOPE - THE CONVENTIONAL CHOICE

For many years, the oscilloscope has been the conventional instrument of choice to produce Lissajous plots. With the advent of multi-channel FFT analyzers, it has become possible to generate Lissajous orbits with only a few problems to overcome. It has been stated that "a well-equipped multichannel FFT analyzer has features which will improve our ability to analyze orbit information, and it can be a more powerful tool than an oscilloscope".[2] Certainly a single channel FFT analyzer can not be used, but a 2-channel FFT analyzer would be adequate for Lissajous orbits. For instance, this is possible by using the "oscilloscope mode" of such a 2-channel analyzer.

While this may be so, for the scope of the material in this chapter, *only the oscilloscope will be featured* in this introductory article.

8.06 CONSIDERATION OF PICKUP LOCATION, DIRECTION OF SHAFT MOTION, AND THE POLARITY OF THE POWER SUPPLY

Nearly all oscilloscopes are designed so that an increase in positive (+) voltage of the input signal (or a decrease in minus (-) voltage) will cause the vertical trace to move *upwards* or the horizontal trace to move to the *right*. (Ref. 2) Two factors need to be considered.

1. The polarity of the non-contact pickup system.
2. The location of the pickup (above or below; on the right or the left side of the shaft).

If the shaft is moving towards a pickup, it is closing the gap, thus reducing the non-contact pickup voltage; if it moves away from the pickup, it is increasing the gap, thus increasing the voltage (i.e., .200 mv/mil gap applied to the DC voltage). *The ideal situation is that in whatever direction the shaft is actually moving with respect to any pickup, the same motion and direction will be duplicated on the CRT.*

When using non-contacting pickups with a power supply with (+) polarity, if the *vertical* pickup is mounted *below* the shaft and the *horizontal* pickup on the *left,* the shaft motion will be read correctly. If they are mounted the other way, *the polarity of the input signal to the scope needs to be reversed to read correctly.*

When using non-contacting pickups with a power supply with (-) polarity, if the vertical pickup is mounted *above* the shaft and the horizontal pickup on the *right,* the shaft motion will be read correctly. If they are mounted the other way, the polarity of the input signal to the scope needs to be reversed.

If this is not carefully done, it will most likely lead to false interpretation of the data. Thus, closely examine the specifications of non-contact pickups installed on machines to be analyzed using orbit pattern analysis.

8.07 SETTING UP THE OSCILLOSCOPE FOR LISSAJOUS ORBITS

Proper calibration of the horizontal and vertical gain controls of the oscilloscope is important to ensure that the shape of the observed pattern accurately reflects the true motion of the shaft, and to make it possible to obtain accurate amplitude data from the scope. While most scopes have separate horizontal and vertical gain controls, using a scope incorporating matched and locked horizontal and vertical gain adjustment can greatly reduce setup and calibration time. In any event, *where system sensitivity is known, calibration of the scope gain settings is simply a matter of setting the oscilloscope gain sensitivities* to match input sensitivities. For example, for a 200 mv/

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mil system, a typical calibration is to adjust the scope to read 200 millivolts per division. In this way, each division on the scope represents 1 mil of displacement. If additional gain is needed, the scope can be set to even submultiples of pickup sensitivity. Thus, for a 200 mv/mil system, setting the gain controls at 100 mv/division would yield 0.5 mil per division or 1.0 mil for 2 divisions. Again, *the horizontal and vertical gain controls must be in agreement so that 1 mil of vertical motion produces the same indication on the CRT as 1 mil of horizontal deflection*. The scaling factor (i.e., mils/division) should be recorded for future reference. (Ref. 2)

Also to enable the analyst to visualize the motion of the shaft in relation to the position of the non-contacting pickups, he may wish to mark lines on the scope screen to represent the actual horizontal and vertical axis of the bearing. Especially if the display was rotated 45° on the CRT (due to the fact that the pickups are 45° from the horizontal on the machine), it would be desirable to do this, as shown in Figure 2.

8.08 INTERPRETING LISSAJOUS ORBITS WITH OR WITHOUT "BLANK" SPOTS

Sometimes Lissajous orbits are shown with "blank spots" superimposed on their orbits. At other times no blank spots or keyphasor marks are shown. The problem is, that *if no blank spots are present at all, it becomes extremely difficult to determine whether a harmonic frequency is predominant or whether a subharmonic is being displayed*. In other words, if no keyphasor reference is used, it will not be possible to determine whether the loops are harmonically or subharmonically related.

In general, if the pattern from 1X RPM is being featured the orbit is known as the displacement orbit. If a trigger, such as a keyphasor is used, it will trigger a once-per-revolution reference pulse which will be superimposed on the displacement orbit as a "blank spot". *If a secondary loop also has a blank spot superimposed on it, the secondary loop represents a subsynchronous component of the rotor's motion. If the blank spot is "locked in-phase" with the blank spot on the displacement orbit, then the subsynchronous component is harmonically related. If the blank spot moves in relation to the blank spot on the displacement orbit, or if the secondary loop moves also, then the secondary loop is not harmonically related; it is nonsynchronous*.

*In general, external secondary loops refer to harmonic orders whereas internal secondary loops represent subharmonic component*s. However, if each of the external secondary loops have a "blank spot", they then represent a particular dominant subharmonic frequency in this case. That is, a "Figure 8" pattern with each orbit having a "blank spot" represents the 1/2X RPM component of the rotor's motion. If there are four orbits with each orbit having a "blank spot", it represents the 1/4X RPM component of the rotor's motion. An excellent example of this is shown in Chapter 5 in Figure 2.

On the other hand, if the orbits represent a particular harmonic multiple, then only the predominant frequency orbit, usually 1X RPM orbit, has the blank spot on it, whereas the other external loops will not have any "blank spots". That is, if the displacement orbit representing the 1X RPM motion has a "blank spot", the external secondary loop represents the 2X RPM component if it remains stationary with respect to the 1X RPM orbit portion. If not, then the external loop is not synchronous with the dominant 1X RPM motion of the shaft. In either case, if the orbit does not have a "blank spot" for higher than 3X RPM components, the orbit becomes complex and beyond the scope of this manual.

In other cases, the orbit may have multiple "blank spots" superimposed on it rather than internal secondary loops, or external loops, each with a "blank spot". *That is, a Lissajous orbit with two "blank spots" means that the 1/2X RPM orbit is being represented; three "blank spots", 1/3X RPM; four "blank spots", 1/4X RPM; and so forth.* It is important to state again that there can be an endless variety of orbits, particularly from rubs, and this chapter is intended to be only introductory to this diagnostic tool.

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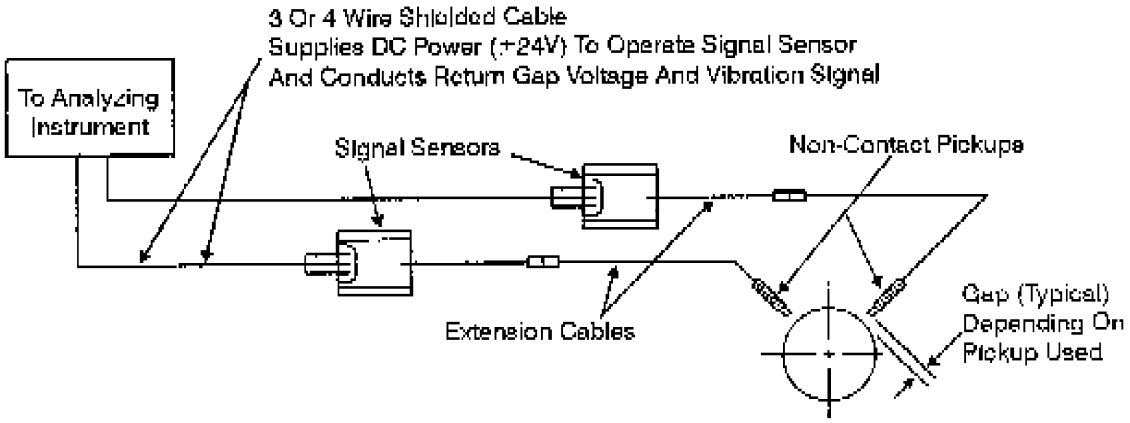
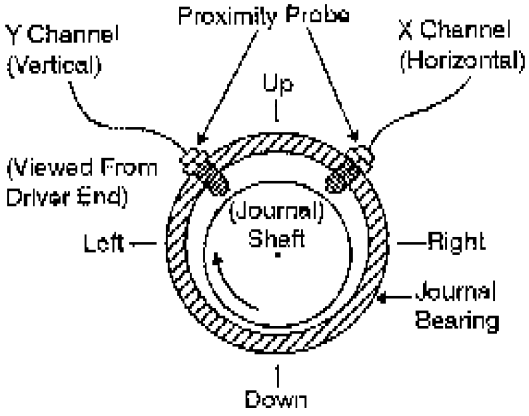
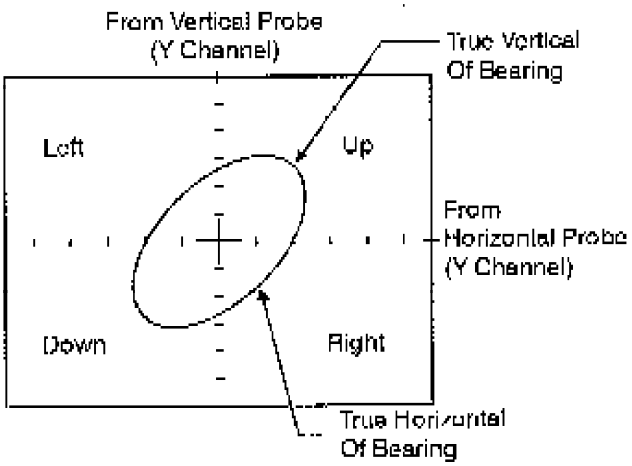

3 Or 4 Wire Shielded Cable
Supplies DC Power (+24V) To Operate Signal Sensor
And Conducts Return Gap Voltage And Vibration Signal
To Analyzing Instrument
Signal Sensors
Non-Contact Pickups
Extension Cables
Gap (Typical) Depending On Pickup Used

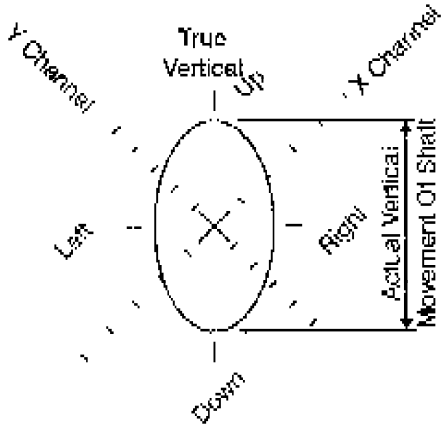
A TYPICAL DUAL PICKUP SETUP USING SIGNAL SENSORS, INSTEAD OF A NON-CONTACT DISPLACEMENT ACCESSORY (SIGNAL CONDITIONER)(Ref. 2)



According to specifications developed by American Petroleum Institute, the position of the probes in the bearing housing should be viewed from the location of the driver. Looking from the driver at the driven, the probe that is on the right hand side should be noted as the horizontal or x-channel probe. The direction of rotation of the shaft is not important in designating which is the "horizontal" and which is the "vertical" probe.



A POSSIBLE LISSAJOUS ORBIT AS DISPLAYED ON AN OSCILLOSCOPE SHOWING THE SCREEEN'S VERTICAL AND HORIZONTAL AXIS OR CHANNELS.



MENTALLY ROTATE THE DISPLAY TO DUPLICATE THE SETUP AT THE BEARING. THE MAJOR AND MINOR AXIS CAN BE MEASURED WITH A SCALE OR THE GRID ON THE CRT.

FIGURE 2
SETUP AND MEASUREMENT OF LISSAJOUS ORBITS (Ref. 3)

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8.1 TYPICAL LISSAJOUS PLOTS FOR COMMON PROBLEMS

The machines to which the analysis of Lissajous plots applies are likely those with journal bearings and non-contact eddy current pickups positioned close to the shaft (usually at a gap of approximately 40 to 60 mils). It is assumed that both pickups have been calibrated and installed properly. If more definition of machine type is required, let the term "machinery" refer to high-speed turbomachinery such as turbines, compressors, and centrifugal pumps. The pickups are installed 90° from each other - one in the horizontal position and the other in the vertical position for simplicity (again; many such probes are oriented at the 10:30 and 1:30 clock positions). The reference pulse giving the once-per-revolution signal is provided by another non-contacting pickup triggering off the key in the shaft. It is assumed that this pickup has also been set up properly. The plots which follow have been modified to simulate the "blank" spot on the orbits which indicates the position on the shaft (via the key).

8.11 UNBALANCE

Ideally, the Lissajous plot for a pure unbalance condition would be a perfect circle. Of course, the diameter of the circle would increase proportional to the increase in operating speed vibration amplitude. There would be one "blank" spot from the reference pulse. Figure 3 is an ideal example of a Lissajous orbit for an unbalance condition.

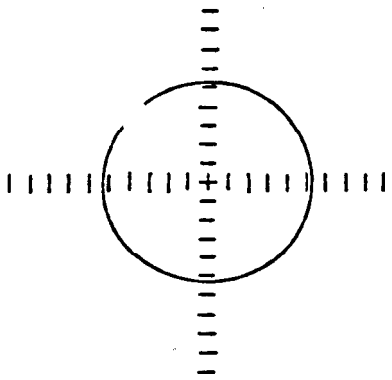

FIGURE 3
EXAMPLE OF A LISSAJOUS ORBIT INDICATING AN UNBALANCE CONDITION (Ref. 5)

Although it is possible for real-world machines to closely approach this ideal condition, *the much more common Lissajous plot indicating unbalance is shown in Figure 4.* To have a perfect circle, the horizontal and vertical stiffness of the bearing would have to be identical. More commonly, machines are stiffer in the vertical (normal; perpendicular) direction because they are bolted to the foundation or base in that direction. The effect of gravity is a force also acting in the vertical (downward) direction. Thus, the Lissajous orbit will be somewhat elliptical in these cases due to unequal stiffness or resistance to motion.

How elliptical can the orbit be before it would no longer indicate an unbalance problem? There are no hard-and-fast rules. In general, if the ratio of the major axis to the minor axis is between 3:1 and 5:1, the Lissajous orbit could still be a result of an unbalance condition. *If the major to minor axis ratio is higher than 5:1, then the condition leans more towards another problem other than unbalance such as misalignment, resonance, eccentricity or excessive clearance due to bearing wear*. Remember, "there is always an exception to the rule".

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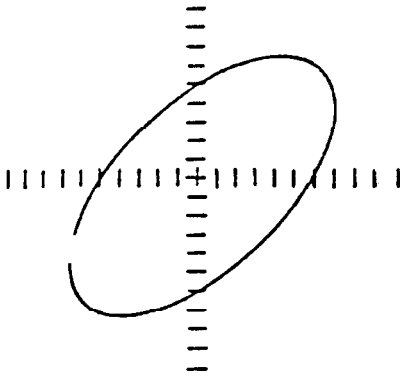

FIGURE 4
AN EXAMPLE OF A MORE LIKELY LISSAJOUS ORBIT INDICATING UNBALANCE (Ref. 5)

8.12 MISALIGNMENT

If there is a shaft misalignment condition, especially across the coupling, the Lissajous plot may resemble an ellipse as shown in Figure 5. In general, the ratio of the major axis to the minor axis is from 3:1 to 8:1 for the elliptical orbit in a misalignment situation, with 5:1 or 6:1 being the norm. However, unlike unbalance, the 1X RPM contribution may be coupled with harmonic orders, 2X, 3X, or higher orders of shaft speed, particularly if mechanical looseness is also involved. Figure 6 is an example of a Lissajous plot from a misalignment situation, possibly coupled with a looseness or shaft "end play" influence. Notice that the ellipse is distorted. It is not a perfect figure eight. The length of the "tail" will indicate the relative amplitude level contributed by the harmonic (distortion) components. If the 1X RPM component is one half or twice the 2X RPM component in amplitude, a "figure 8" orbit as shown in Figure 7 may be formed. In this case also, the pattern and the "tail" will be influenced by the phase and the relative amplitudes of any other harmonic components which may be present.

In cases of excessive misalignment, other shapes are possible. There may be a truncated orbit, shown in Figure 8(a); a banana-shaped orbit, shown in Figure 8(b); and others, as shown in Figure 8(c). These more complex shape examples result from the distortion of the 1X RPM displacement component with those from the higher harmonic orders.

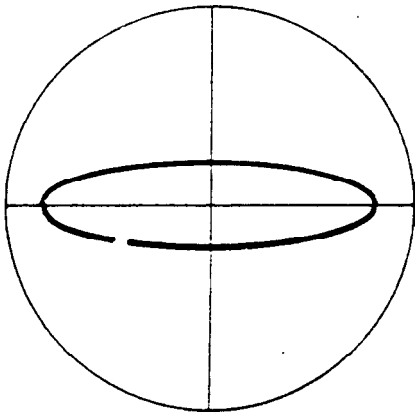

FIGURE 5
A LISSAJOUS ORBIT EXAMPLE FOR A MISALIGNMENT PROBLEM (Ref. 2)

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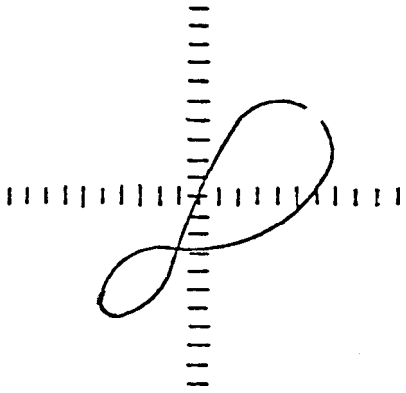


FIGURE 6
A "FIGURE 8" LISSAJOUS ORBIT EXAMPLE FOR A COMBINED MISALIGNMENT AND LOOSENESS PROBLEM (Ref. 5)

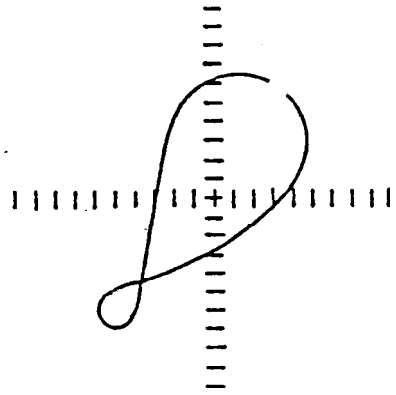
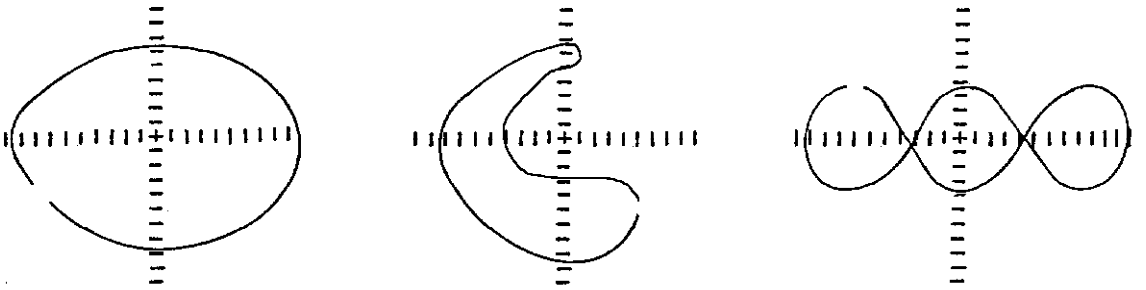


FIGURE 7
AN EXAMPLE "FIGURE 8" LISSAJOUS ORBIT FOR A MISALIGNMENT PROBLEM WHERE 1X RPM & 2X RPM HAVE A 2:1 RATIO (Ref. 5)



(a) A TRUNCATED ORBIT (b) A BANANA-SHAPED ORBIT (c) A COMPLEX-SHAPED ORBIT

FIGURE 8
VARIOUS EXAMPLES OF LISSAJOUS ORBITS FROM CASES OF EXCESSIVE MISALIGNMENT

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8.13 HOW CAN IT BE DETERMINED WHETHER THE LISSAJOUS ORBIT IS CAUSED BY UNBALANCE, MISALIGNMENT OR RESONANCE?

There is no "surefire" way to differentiate between these problems with orbit analysis alone. A key to the solution would be the ability to change the shaft speed higher or lower. Another key would be to watch the change in position of the reference "blank" spot as the speed is changed. For example, to determine *if the orbit is caused by unbalance or resonance, (particularly if the shape is elliptical), watch the position change (orientation) of the "blank" spot*. If it is assumed that the shaft is running below its first critical speed and the shaft RPM is increased, the "blank" spot on the orbit will shift as shown in Figure 9. At resonance the "blank" spot will shift approximately 90° away from where it was before the shaft speed was changed. If the speed continues to increase past the first critical, the "blank" spot will shift approximately 180° away from its original location as shown in Figure 9d and 9e. However, it was only assumed that the shaft was operating below its first critical. If the shaft speed is located between the first and second critical, a similar position change of 180° in the "blank" spot could be observed as the shift decreases in speed to below its first critical. Then, it could be concluded in both cases, that the elliptical shape was due to the shaft speed being in close proximity to its critical speed. It would not be due primarily to an unbalance condition. If the "blank" spot does not move as the speed is increased, then it could be concluded that the problem is from unbalance or possibly from misalignment. Also, the shape of the ellipse will stay basically the same with unbalance but the major and minor axis will increase proportional to the increase in vibration amplitude.

To determine whether the elliptical orbit is from misalignment or resonance, observe the change in the shape of the orbit as the speed is varied. If the shaft is running near its critical speed, then the shape of the Lissajous plot will change noticeably if the shaft speed is increased or decreased as shown in Figure 9c. *If the ratio of the major axis to the minor axis is 8:1 to 10:1, or more, this strongly suggests that the rotor is operating near its critical speed or even possibly near a structural resonance*. Once again, watch for the change in shaft speed to affect the shape of the Lissajous orbit. (Ref. 2)

These suggestions are only meant to be guidelines, not hard-and-fast rules. Machinery dynamic behavior is very complex; often, solutions are not this simplistic. For instance, in many cases it is not an easy process to increase or decrease the shaft speed. If this is not expedient or practical, *a judgement call is often made based on the available diagnostic information on hand.*

8.14 ROTOR RUBS

If a machine is subjected to a rotor rub, it has no internal corrective responses with which to eliminate the effects of the rub. A rub is generally caused when the rotor contacts a stationary component, such as the bearing surface, a seal, and in the case of a motor, the stator. There are many causes of rubs with varied effects. Some common causes are: excessive unbalance, misalignment, soft foot, excessive clearance in the bearing, thermal bow, and others. Rotor rubs are basically one of two types: partial rubs and full rubs.

8.141 Very Mild Rotor Rubs

A very mild rotor rub is one in which the shaft merely "touches" the stationary component once per shaft revolution. The result is a slight distortion of a circular and elliptical orbit as shown in Figure 10. Along with this slight distortion, there will be a back-and-forth movement of the "blank" spot in the Lissajous orbit. *This movement of the "blank" spot representing a keyphasor oscillation is called keyphasor "jitter"*, particularly when the position of the "blank" spot changes by more than 10°. However, mild rubs often deteriorate into full rubs over time. (Ref. 4)

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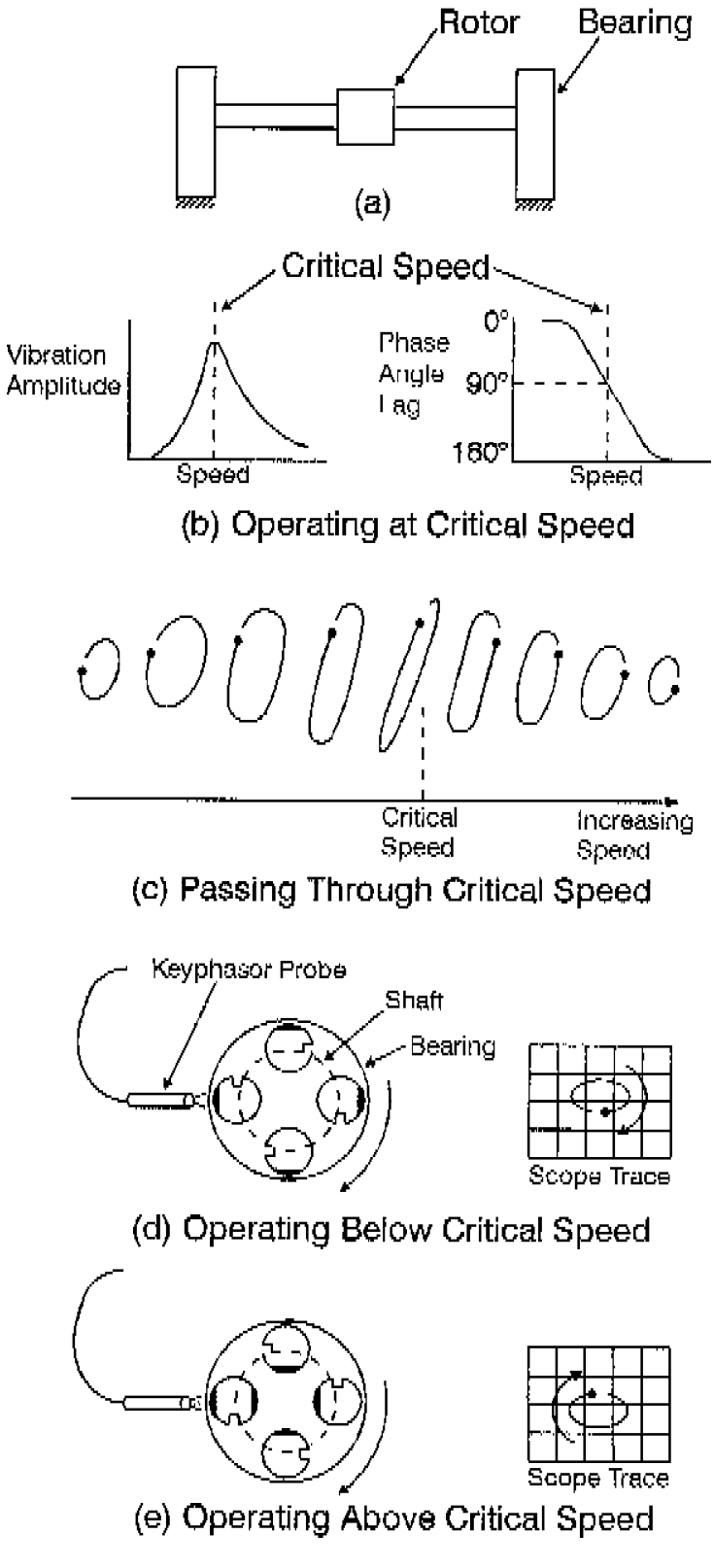

FIGURE 9
OPERATIONAL CHARACTERISTICS OF A SIMPLE ROTOR SYSTEM (d) BELOW AND (e) ABOVE THE FIRST CRITICAL SHOWING THE EFFECT ON LISSAJOUS ORBITS

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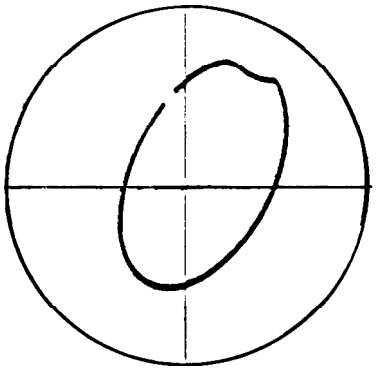

FIGURE 10
AN EXAMPLE OF A LISSAJOUS ORBIT FROM A VERY MILD ROTOR RUB
(Ref. 2)

8.142 Heavy or Full Rubs

When the rub becomes more severe, the Lissajous orbit tends to become increasingly more complex and erratic. *Component resonances can be excited, increasing orders of harmonic frequencies can appear, and other nonsynchronous frequencies interact to create an endless number of possible effects*. Figure 11 is a representative example of various patterns of Lissajous orbits for heavy or full rubs.

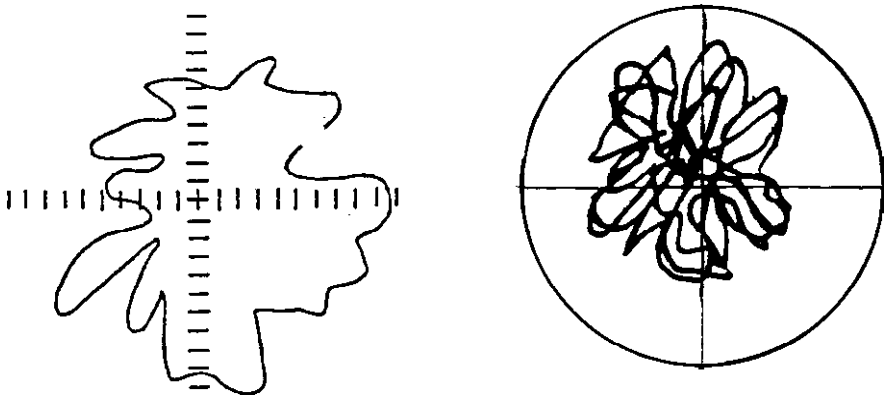

FIGURE 11
EXAMPLES OF LISSAJOUS ORBITS FOR HEAVY OR FULL ROTOR RUBS
(Ref. 2, 4)

By observing the oscillations of the "blank" spot, an insight into the severity of the rub can be obtained. *As the rub degenerates into a worse condition, the movement of the "blank" spot will become more random and erratic*. It is a reflection of what is occurring to the shaft (via the key) as the rub takes place. Initially, with a slight rub, there is a slight resistance to the direction of shaft rotation. As the amplitude level increases because the situation is getting worse, there is a tangential force which acts on the shaft where it contacts the bearing or seal in a direction opposite to shaft rotation. At the time the shaft actually contacts the stationary surface increases, a reverse precession will be observed in the Lissajous orbit. When the tangential force becomes large (as in the case of a full rub), *the shaft will orbit in a direction opposite to the direction in which it is driven to rotate*. In most cases, a shaft which is experiencing a full rub condition will lead to a catastrophic failure of the machine. Figure 12 is an example orbit of this reverse precession during a full rub. The presence of 4 "blank" spots indicates that the frequency of the orbit is 1/4 X RPM.

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FIGURE 12
AN EXAMPLE LISSAJOUS ORBIT OF THE REVERSE PRECESSION IN A FULL ROTOR RUB (Ref. 4)

8.143 "Hit and Bounce" Rubs

Between the extremes of a mild rub and a full rub is a family of partial rubs, known as the "hit-and-bounce" variety. A condition such as misalignment can cause a shaft to lightly touch the bearing surface which suppresses the rotor's tendency to move outward radially. However, as the condition degenerates, more happens to the orbit other than a mere flattening at one point-per-shaft revolution. The shaft will hit and bounce off. This can result in a reverse whirl. Most often this occurs at exactly 1/2X RPM. Reference 2 provides the following helpful information: *"two full revolutions of the shaft have to be set up in the instrument to be able to display this Lissajous orbit"*.[1] This will be shown in the orbit as a secondary *internal* loop. In another setup, two stationary "blank" spots may be present on a single orbit. This will indicate that the frequency is 1/2 the shaft RPM (Ref.2). As the condition degenerates, more and more hits can occur and they will be reflected in the Lissajous orbits, as shown in Figure 13. Notice that as the hits become more frequent, the internal loops become tighter and tighter.

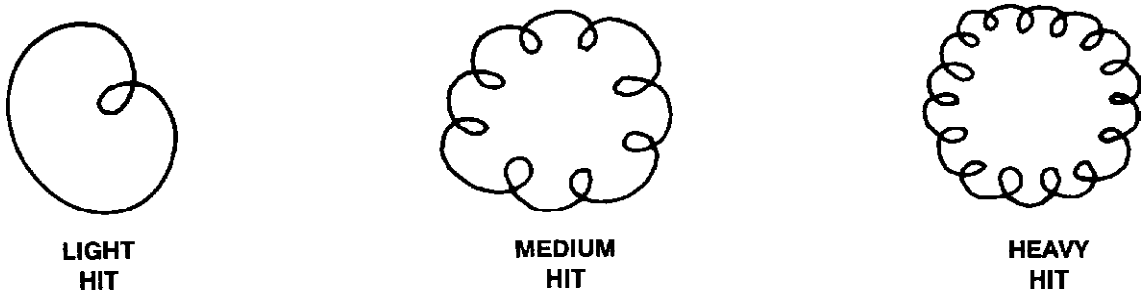

FIGURE 13
EXAMPLE LISSAJOUS PLOTS OF VARIOUS "HIT-AND-BOUNCE" ROTOR RUBS (Ref. 4)

8.144 Conclusions From Lissajous Orbits in Rotor Rub Diagnostics (Ref. 3)

For partial rubs:
1. They create orbits with internal loops.
2. They cause the "blank" spot to jitter.

For full rubs:
1. The orbit rotation is *opposite* the direction of shaft rotation (reverse precession process).
2. Even though the shaft may slow down due to the friction in the rub, the frequency the reverse rub orbit is virtually constant.

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8.15 OIL WHIRL

Oil whirl usually occurs in machines with journal bearings and a pressurized lubrication system. Generally, the shafts are long and have relatively high RPM's (operate above their first critical). The bearings are lightly loaded with low damping. Unbalance or misalignment usually provides the excitation force that supplies the impetus behind oil whirl.

The shaft has a tendency to ride on wedge of oil which has a lifting effect on the shaft. This "small destabilizing force [combines with the excitation force to] cause the shaft and lubricant to circulate strongly in the direction of the shaft rotation",[1] which causes the shaft to whirl as it rotates. *The oil whirl frequency will be between 40% and 48% of the shaft speed.*

The Lissajous orbit for oil whirl condition will show a secondary internal loop that slowly rotates about the axis of the displacement orbit. It rotates at a frequency independent of the displacement orbit because it is *not synchronous with rotation*. Figure 14 is an example of a Lissajous orbit from a shaft subjected to an oil whirl condition. (Ref. 2)

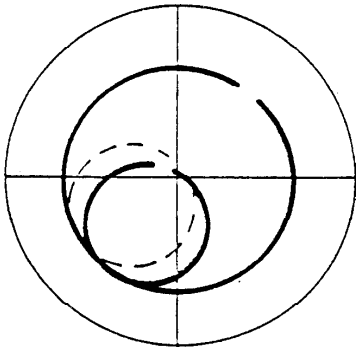

FIGURE 14
AN EXAMPLE LISSAJOUS ORBIT FROM AN OIL WHIRL CONDITION (Ref. 2)
(Internal Loop Slowly Rotates About the Axis of the Orbit)

8.16 MECHANICAL LOOSENESS

The Lissajous orbit behavior due to mechanical looseness is similar to that for a rub. The most common cause of mechanical looseness is too much clearance in the bearing. *The outstanding effect of mechanical looseness is the presence of integer fraction subharmonics of 1X RPM, (usually 1/2X or 1/3X), as well as the presence of harmonic orders of 1X RPM of the shaft. The subsynchronous frequencies will produce secondary internal loops similar to the Lissajous orbits for rotor rubs.* Of course, in more complex situations, the orbits will be quite erratic, as shown in Figure 15.

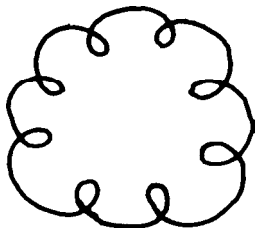

FIGURE 15
AN EXAMPLE LISSAJOUS ORBIT SHOWING COMPLEX INTERNAL LOOPS DUE TO MECHANICAL LOOSENESS

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8.17 HOW CAN IT BE DETERMINED WHETHER THE LISSAJOUS ORBIT IS FROM MECHANICAL LOOSENESS, ROTOR RUB OR OIL WHIRL?

To determine whether the problem is from mechanical looseness or a rotor rub, the key to the solution is in determining the direction that the displacement orbit moves in relation to the shaft rotation. If the problem is mechanical looseness, the displacement orbits and secondary loops will be locked "in-phase" regardless of the direction the shaft is rotating. *If the problem is rotor rub, the displacement orbit will have a rotating tendency* (via the "blank" spot) *in a direction opposite to shaft rotation, especially in full rub situations.*

To determine whether the problem is from mechanical looseness or from oil whirl, consider the secondary loop. *If the problem is oil whirl, the internal loop will slowly rotate in the same direction as shaft rotation.* If the problem is mechanical looseness, the secondary loops will be stationary since they will be synchronous with the 1X RPM displacement orbit. This "in-phase" feature will also apply to rotor rub.

8.2 APPLICATIONS OF LISSAJOUS ORBITS NOT COVERED

There are more applications and diagnostic features of Lissajous orbits which are beyond the scope of this chapter. It was intended to be an introduction to the use of Lissajous orbits for machinery diagnostics. Lissajous orbits are also used to measure integral ratios between sine waves of different frequencies. The examples and Lissajous orbits used in the chapter were basic. In the real world, the non-sinusoidal effects from gear imperfections, bearing faults, electrical vibrations and so forth, will cause "hash" on the waveforms. One can imagine the effects on the Lissajous orbits. In these cases, sophisticated instrumentation and persons with great expertise may be required to separate the real problems from all the background "noise", as well as to evaluate the severity of problems found.

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3. Wonk, Victor; *Machinery Vibration: Measurement and Analysis*; McGraw-Hill, Inc.; 1991, pg. 260.

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CHAPTER 9

ROLE OF SPIKE ENERGY, HFD AND SHOCK PULSE (SPM) & SPECIFICATION OF THEIR ALARM LEVELS AT VARIOUS SPEEDS

9.1 SPIKE ENERGY AND SHOCK PULSE:

Ultrasonic instruments are designed to measure energy levels in roughly the 25,000 Hz to 100,000 Hz frequency range (1,500,000 - 6,000,000 CPM). These include *Spike Energy*[1] and *Shock Pulse*[2]. Each of these two parameters were developed in the 1970's and are designed to mechanically and electrically respond only to high frequency problem sources which excite the resonant frequency of their mounted transducer. Note the effective range for Spike Energy is from 5,000 Hz to 60,000 Hz as shown in Illustration A. In the case of spike energy, one of the most common accelerometers now in use (*IRD 970) has a mounted natural frequency of about 30,000 Hz*. Similarly, the reference mass within *the shock pulse transducer* is designed to respond at its *resonant frequency of approximately 32,000 Hz*.

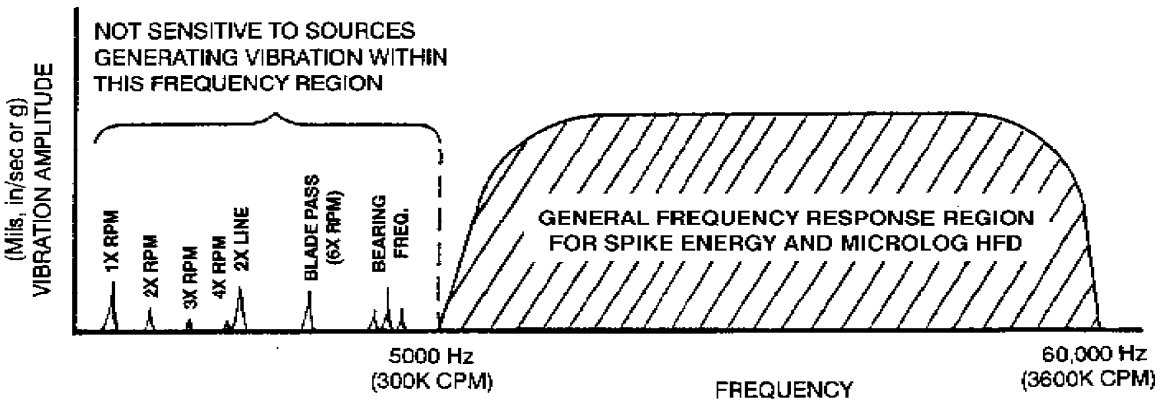

ILLUSTRATION A.
GENERAL FREQUENCY RESPONSE CURVE
FOR SPIKE ENERGY AND MICROLOG HFD

Each of these technologies has proven worthwhile and capable of acting as effective tools at picking up initial stages of bearing wear *if the user follows very strict measurement rules* (these are covered in a Vibration Institute article entitled "The Use of Spike Energy for Fault Analysis and Machine Condition Monitoring" written by Joseph M. Shea and James K. Taylor of IRD). *When bearings first begin to wear*, they begin to excite natural frequencies of their components themselves (rolling elements, bearing races and cage) as discussed in the special rolling element bearing section in Chapter 6, Section 6.09. One set of bearing component natural frequencies is concentrated within the range of 500 to 2000 Hz (30,000 to 120,000 CPM). *Another set is found within ultrasonic frequency ranges near the SPM and spike energy transducer natural frequencies.* Therefore, when incipient wear just begins within rolling element bearings, the bearing components begin to impact, exciting these natural frequencies which likewise excite the SPM and spike energy transducer natural frequencies. Basic defect mechanisms which generate ultrasonic response include:

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a. Micro spalls and cracks from fatigue or overstressing caused by brinelling, false brinelling, misalignment, overload, incorrect sealing and improper fits.

b. Surface roughening from lack of sufficient lubrication.

c. Surface indentations from hard contaminant micropitting from electric current passing through the bearing.

In the case of spike energy, *accelerometers have been designed which have mounted natural frequencies on the order of 30,000 Hz (1,800,000 CPM) in the ultrasonic range*. The sharp pulses and broadband random ultrasonic excitation from the bearings will excite the accelerometer natural frequency whether the accelerometer is stud-mounted, magnet-mounted, or probe-mounted (obviously, the stud-mount gives the highest spike energy *magnitudes* and most repeatable results). But all three mountings will provide spike energy measurements. This is somewhat surprising when one recalls what happens to *vibration* measurement quality results when he goes from stud to magnet to probe mount. However, like the empirical data that has been captured on rolling element bearings showing that their set of ultrasonic natural frequencies shows little change whether "free-free" or with the bearings mounted in the machine, the same appears to be true in the case of spike energy. *In "layman's" terms, spike energy is generated when the incoming ultrasonic frequencies excite the accelerometer natural frequency, an electrical voltage response occurs which is then electronically conditioned and filtered, converting the end product to what might be called "impact energy" which is proportional to the incoming impact velocity. It is probably best thought of as a friction measuring parameter*. Figure 1 shows a spike energy time waveform which would be seen by measurement on a faulty rolling element bearing.

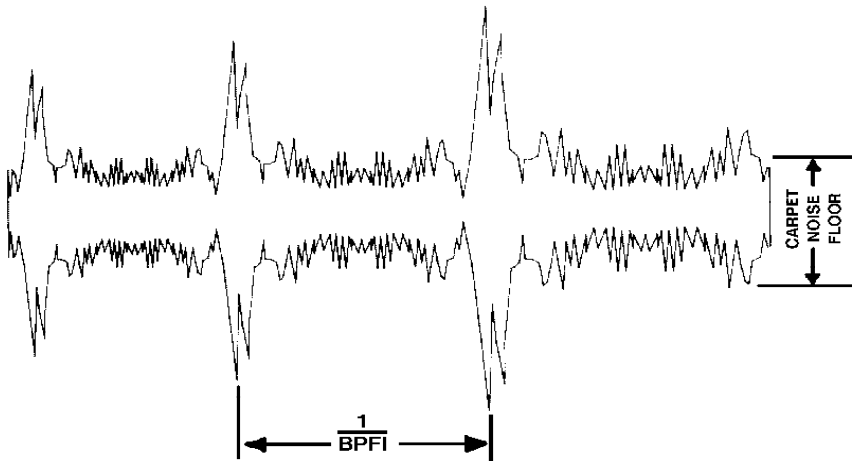

FIGURE 1
SPIKE ENERGY TIME WAVEFORM FOR A FAULTY ROLLING ELEMENT BEARING

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These pulses acting at high frequencies propagate rapidly through the structure. In the case of bearings, these elastic wave energy pulses travel through the bearing and surrounding structure at velocities approaching 16,000 ft/sec which is roughly the velocity of sound (elastic) waves in steel (10,900 mi/hr!). *When these impact waves encounter an interface of two materials or components, they lose much of their energy*. This is shown in Figure 2. This interface may be a region between the bearing outer race and its housing or between one machine part and another to which this part is fastened. When these ultrasonic waves encounter an interface, some of the energy is transmitted through, and some is reflected back towards the source. The amount of energy reflected depends on a number of factors: the sharpness of the pulse, the difference in metals at the interface, how tightly the sources mate together, etc. In general, *it is estimated that typically approximately 60% to 80% of the energy is reflected at each interface*. **THEREFORE, THIS EXPLAINS WHY IT IS SO IMPORTANT TO KEEP THE MEASURING ULTRASONIC TRANSDUCER AS CLOSE AS POSSIBLE TO THE BEARING BEING EVALUATED.**

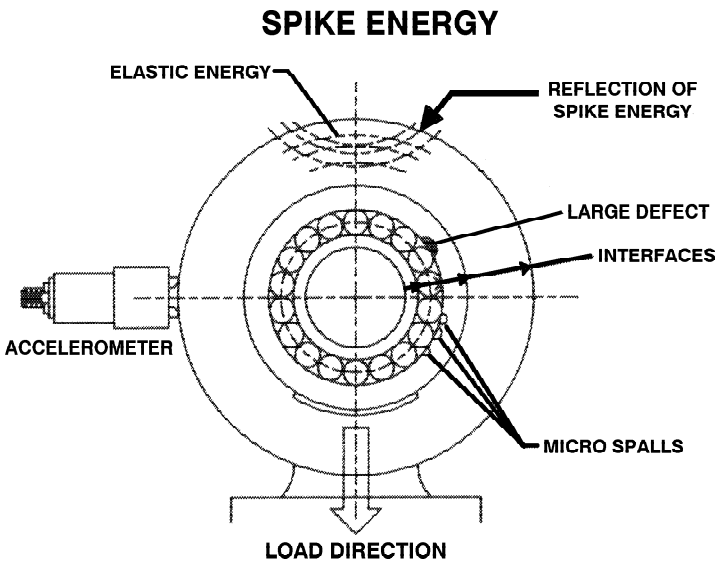

FIGURE 2
REFLECTION OF SPIKE ENERGY AT COMPONENT INTERFACES

While this energy loss may be considered as a weakness, it also has an advantage. For example, when a bearing begins to fail, much of its vibration not only will exist within this bearing, but will transmit to other structures and bearings. On the other hand, ultrasonic energy is much more localized, making it easier to isolate the specific bearing having the problem.

Here, it is important to point out that not only does spike energy and shock pulse respond to bearing wear, but also such *ultrasonic measurements respond to each of the following:*

- bearing wear (as stated before)
- bearing lubrication
- cavitation
- rotor or seal rub
- belt squeal
- gear meshing

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- sheave rub against a guard
- impact excitation of machine parts (i.e., valve in a reciprocating machine)
- steam and high pressure air flow

Since they do respond to all of these problems, this makes ultrasonic measurements of even greater use. In some cases, it might be confusing whether the problem is radiating from a bearing or from one of the other many sources listed above. *However, if one remembers that ultrasonic energy dissipates rapidly (is very localized), it can help him in diagnosing which of several problems might exist*. For example, in the case of a pump, if spike energy is high on both pump bearings, it is very likely that the pump might be experiencing cavitation or a seal rub. If the operator likewise records that he hears something like a "gravel sound", it might make a strong case for cavitation. On the other hand, if on the same pump, spike energy were high only on one of the two bearings, it might suggest possible wear or lubrication problems with this one bearing.

Finally, it is also important to point out ***what ultrasonic measurements do not respond to***:

- unbalance
- misalignment
- bent shaft
- electrical problems
- eccentric rotors
- resonance
- structural looseness/weakness
- beat vibration problems

The reason that neither shock pulse nor spike energy respond to the above problems is that they generate lower frequency vibration which is purposely filtered out in the design of their electronics [neither SPM nor overall spike energy are sensitive to problems which generate fault frequencies below about 5000 Hz (300,000 CPM)]. Still, they are very useful in picking up the other problems such as bearing, lubrication, cavitation and rub problems.

Each of these technologies likewise have some disadvantages. In the case of shock pulse, the bearing bore and RPM (or bearing type and number) *must* be known in order to take its reference baseline (otherwise, the results are inaccurate). This is a disadvantage on many machines whose bearing model numbers and bores are not known. Also, since the shock pulse instrument usually makes only ultrasonic measurements, a second instrument is required to obtain vibration readings (particularly *filtered* vibration spectra) which will detect the lower frequency problems. In addition, since the shock pulse instrument is not yet included within any of the major predictive maintenance software offerings, its readings have to be manually entered into these programs if one wants to both store and trend these measurements in his PMP software allowing him to directly compare vibration and shock pulse measurements in one trend graph. (However, SPM does now offer its own software for trending shock pulse readings themselves).

In the case of spike energy, one of its disadvantages currently is that its amplitude response is highly dependent on which specific accelerometer (and mounting) is used to make the measurement. Amplitudes may differ by a factor of 5X to 10X from one accelerometer to another. Thus, if one wants to use current spike energy severity charts like that shown in

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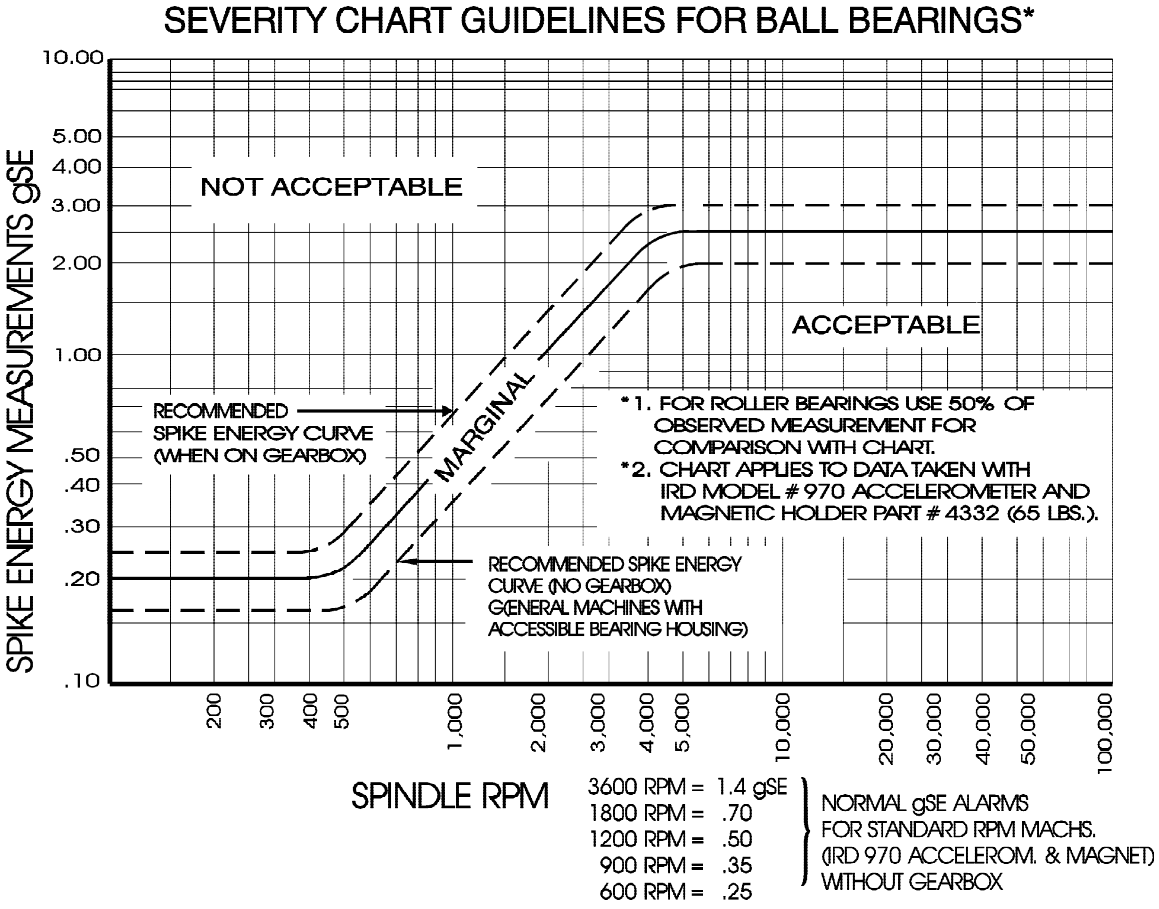
FIGURE 3
RECOMMENDED SPIKE ENERGY SEVERITY CHART (IRD SPIKE ENERGY)



SPIKE ENERGY™ MEASUREMENTS

Energy is generated by repetitive transient mechanical impacts. Such impacts typically occur as a result of surface flaws in rolling-element bearings or gear teeth.

This energy is conducted from its source through various paths to the outer surface of the machine structure, and is seen as a small-amplitude vibration at the surface. Accelerometers coupled to the surface generate corresponding electrical signal.

The accelerometer signals processed by unique filtering and detection circuitry to produce a single "figure of merit" related to the intensity of the original impacts. This figure of merit is expressed in "gSE" units.

SPIKE ENERGY™ gSE readings are measurements which can with experience, be correlated with the severity of the casual surface flaws. Even though gSE readings are affected by the nature of the conductive path between the impact source and the accelerometer, similar machine structures will provide a reasonable basis for comparison between the structures.

The gSE figure of merit has proven to be effective in detecting mechanical defects in meshing gears and rolling element bearings. The gSE measurement, when used in conjunction with conventional measurement of vibration velocity and acceleration, provides early indications of mechanical deterioration.

** When used with magnetic holders, accelerometers must be installed with a light coating of silicone grease and tightened to 40 in-lb. torque.

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Figure 3, this chart mandates use of the IRD model # 970 accelerometer and model #4332 magnet (65 lbs) to determine whether or not the gSE readings are acceptable. At present, readings from any other accelerometer cannot be used with Figure 3 although work is underway for deriving similar charts for other accelerometer/magnet combinations. One advantage spike energy has over SPM is that one does not have to know the bearing bore and model number to take a reading. However, another disadvantage has been found that when spike energy measurements are taken in the presence of steam, the steam itself can cause variant readings.

The advantages for *both shock pulse and spike energy readings depend on the fact that they mandate that one always takes measurements at identical points from one survey to the next using the same model accelerometer and mounting* (not doing so will cause trends to vary widely). This will improve the accuracy, reliability and repeatability of the data from one survey to the next. In addition, if readings are taken with hand-held probes rather than magnet or stud mounted, simply a change in force on the transducer by the operator can raise or lower the subsequent reading. In both cases, if readings can be taken from stud-mounted locations, trends can greatly be improved. Spike energy measurements have also proven fairly repeatable using magnet mounts (particularly if a high force magnet is mounted on a uniform surface), though not up to the quality and repeatability of those from stud mounts.

FIGURE 4
MAINTENANCE DIAGNOSTIC VIBRATION AND FILTERED HIGH FREQUENCY GENERAL TOLERANCE CHART FOR PROCESS MACHINERY WITH ROLLING ELEMENT BEARINGS①

Unfiltered Vibration Velocity	Shock Pulse & Acoustic Emission	Microlog HFD	IRD Spike Energy②	Severity Level and Alarm Status
Inches/Second Peak	dB	g	gSE	
1.5+	50+	5.0+	3.0+	Danger (Shutdown)
.75 - 1.49	40 - 49	3.0 - 4.99	1.50 - 2.99	Very Rough (Alert)
.40 - .74	30 - 39	1.50 - 2.99	.80 - 1.49	Rough (Alert)
.20 - .39	20 - 29	.75 - 1.49	.40 - .79	Fair (Acceptable)
.10 - .19	10 - 19	.30 - .74	.20 - .39	Good (Acceptable)
.01 - .09	1 - 9	.01 - .29	.01 - .19	Smooth (Acceptable)

① Provided by Charles Berggren of Monsanto
② Spike Energy Amplitudes measured using an IRD 970 accelerometer outfitted with IRD's 2-pole, 65 lb. magnet.

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In any case, ultrasonic measurements are recommended on high-speed machines or lower speed machines having high frequency vibration sources (rolling element bearings, gears, etc.). *They can prove very effective in picking up many problems that might otherwise go unnoticed, particularly during incipient stages*, but sometimes even in more advanced stages. In the case of bearings, each of these tools will pick up problems during the first of four definable stages of degradation. Finally, meaningful severity charts are now being developed for each of these high frequency measurement parameters. Figures 3 and 4 are included as example severity charts for spike energy and shock pulse, respectively.

9.2 HIGH-FREQUENCY ACCELERATION (HFD):

Some vendors have offered banded high-frequency acceleration measurements in addition to vibration with their predictive maintenance hardware and software systems. While this is similar to the ultrasonic measurements mentioned above, this is truly a different measurement parameter altogether. In fact, as the name implies, this is a banded high frequency acceleration measurement which measures the total energy existing between certain lower and higher cutoff frequencies. Most often, this high-frequency measurement is referred to as *"HFD*[3]" and is marketed by some of the major predictive maintenance vendors (see "Endnotes" 3 and 4). Figure 4 shows severity levels for "Microlog HFD" as compared with those of spike energy and shock pulse. In general, note that Microlog HFD levels tended to be approximately 2X those of spike energy (that is, about 100% higher in this particular study).

The thing which differentiates HFD measurements from those of shock pulse and spike energy is that HFD has not generally been an ultrasonic measurement, but instead a banded acceleration measurement. In the beginning, HFD by Vendor 3 was normally banded between approximately 5000 Hz and 20,000 Hz (300,000 - 1,200,000 CPM). In recent times, this vendor has allowed the user to specify the HFD lower and higher cutoff frequencies up to as high as 20,000 Hz (1,200,000 CPM). In these cases, it is understood that some users have specified their lower cutoff frequency as low as 1000 Hz up to 5000 Hz and their higher cutoff frequency anywhere from 3000 Hz to as high as 20,000 Hz. "Microlog HFD" (by Vendor 4) provides a fixed frequency range of 5000 Hz to 60,000 Hz (300,000 CPM - 3,600,000 CPM). Like spike energy and shock pulse, HFD is sensitive to faults generating high frequencies such as bearing wear, gear wear, cavitation, etc. Due to its high frequency banding, HFD is not sensitive to such problems as unbalance, misalignment, eccentricity, etc.

Here, it will be of great importance that the analyst consciously uses the same analyzer, transducer and transducer mount (hopefully magnet, quick lock or stud/adhesive mount) in order to make his HFD measurement more repeatable and more reliable.

ENDNOTES:

1. "*Spike Energy*" - by Entek IRD International Corp. based in Cincinnati, OH.
2. "*Shock Pulse*" - by SPM Instruments based in Marlborough, CT.
3. "*HFD*" - by CSI based in Knoxville, TN.
4. "*Microlog HFD*" - by SKF Condition Monitoring based in San Diego, CA.

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9.3 SPIKE ENERGY MEASUREMENTS

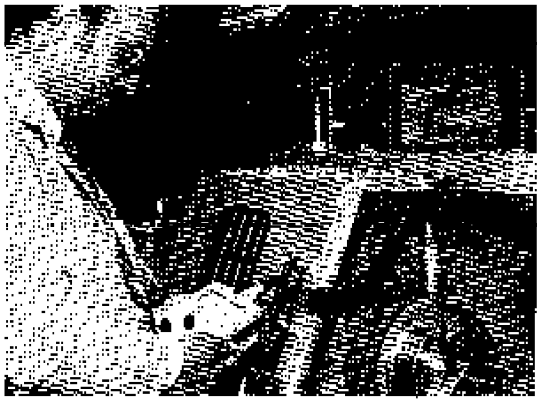
Spike Energy has been defined as "vibration energy generated by short-duration, metal-to-metal impacts and random vibration that propagate through the structure". The following discussion shows how Spike Energy measurements are useful for evaluating the condition of rolling element bearings.

Bearings are one of the most precisely made devices available, so it follows that the characteristic vibration level of a good bearing is quite low compared to other parts of a machine. During the early stages in the development of a bearing defect, the bearing vibration might still be so low that it is lost in the vibration caused by other parts.

The important point of this discussion is that overall filter-out measurements of vibration velocity or acceleration often do not show the characteristics of a bearing defect until the condition reaches a critical point. So measuring or monitoring the overall vibration might not provide sufficient warning of developing bearing problems.

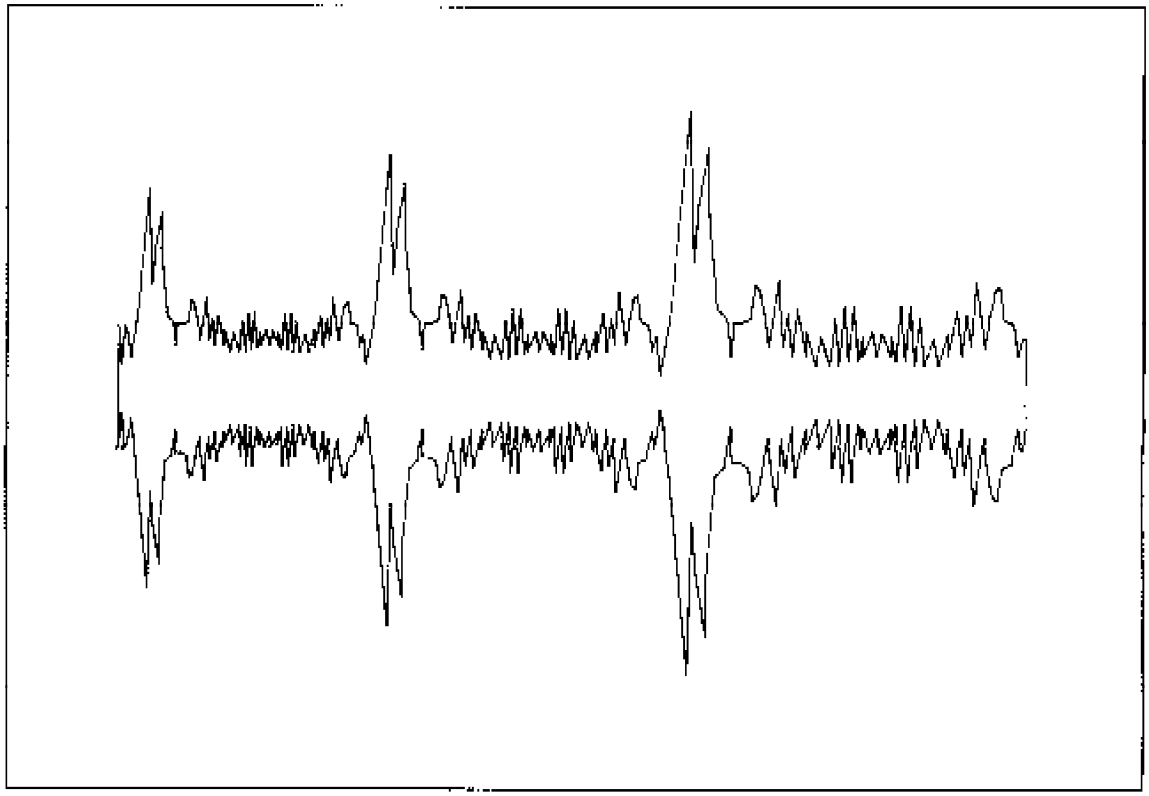
Figure 6-25 shows an IRD Model 810 Vibration-Spike Energy Detector. This instrument includes provisions for measuring normal vibration displacement, velocity and acceleration. It also includes special provision for measuring Spike Energy levels - vibration that is characteristic of defective bearings and gears.

Figure 6-25
An IRD Model 810 Vibration Meter



As illustrated in Figure 6-26, a defect on the bearing raceways or rolling elements causes intermittent impacts between the bearing components. These brief impacts, in turn, excite the resonant frequencies of the various bearing components. A number of IRD instruments respond to these characteristics of bearing vibration, providing information that is needed for assessing bearing condition.

Figure 6-26
Spike Energy pattern due to a defective bearing



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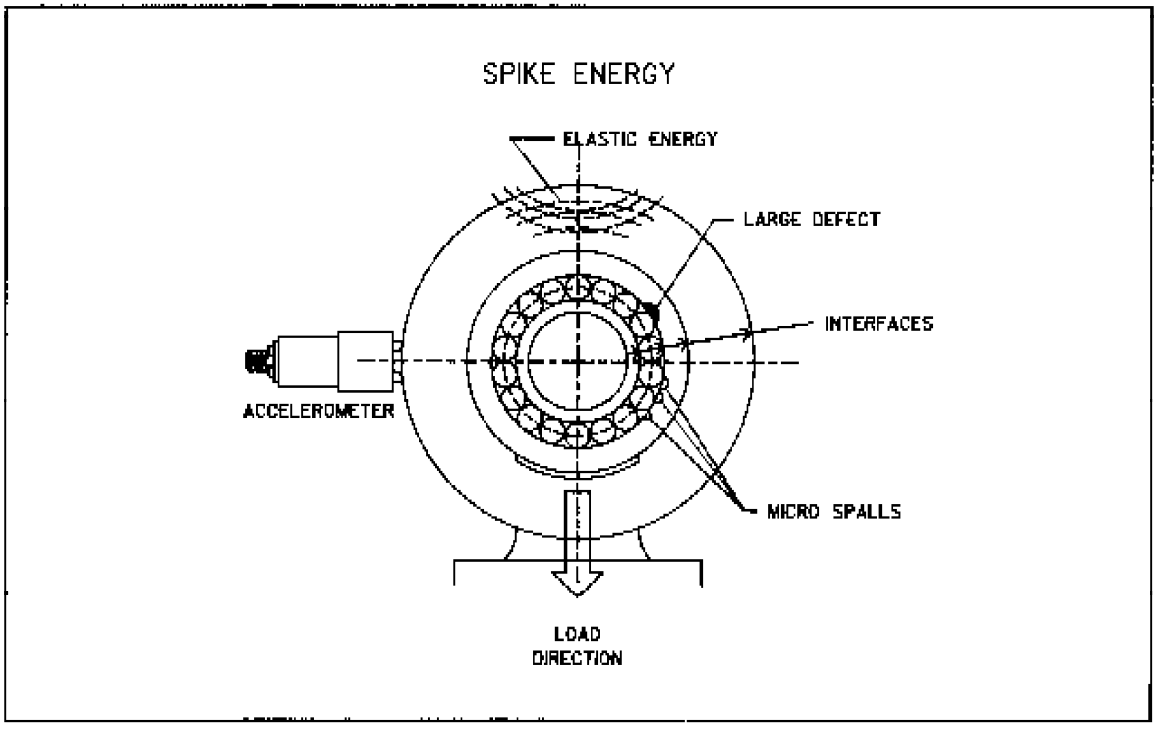
An IRD Model 970 accelerometer pickup has a mounted natural frequency of about 27,000 Hz, or 1,620,000 CPM. This frequency is well above the vibration frequencies generated by mechanical problems in the machine: unbalance, misalignment, electrical problems, aerodynamic forces, hydraulic forces and so on. As a result, the only sources of vibration that excite the natural frequency of this accelerometer are the impact, or spike, forces generated by a defective bearing or gear tooth.

Furthermore, setting the vibration meter to measure Spike Energy introduces a high pass filter that excludes any remaining frequency components below 5000 Hz (300,000 CPM). Consequently, increases in vibration due to problems such as machinery unbalance and misalignment do not cause any increase in the Spike Energy readings. In other words, the Spike Energy modes of IRD instruments are designed to respond to impacts and random vibration forces that are characteristic of defective bearings.

Figure 6-27 shows an accelerometer attached to a machine surface. The energy due to bearing impacts is conducted through various paths to the outer surfaces of the machine. This signal at the accelerometer pickup has a very low amplitude, but special signal-processing circuits in Spike Energy instruments are capable of converting the signal into a numerical quantity, or figure of merit. This value is expressed in g SE units.

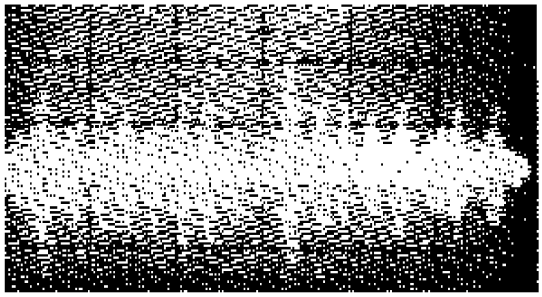
Figure 6-27 Bearing defects generate vibration (SPIKE ENERGY)



Observing the bearing vibration as a time waveform on an oscilloscope can also be helpful in detecting specific bearing frequencies. The time waveform in Figure 6-28 was obtained from a bearing having a known defect on the inner raceway. As the rolling elements pass over the defect, they create the spikes that you can clearly see in the display; and because you note the period of the display, you can readily determine the frequency of vibration. Having determined the frequency of the vibration in this fashion, you can compare it with calculated rotational frequencies of the bearing.

The vibration signal shown on the oscilloscope display varies in amplitude due to the fact that the defect on the inner race is rotating, and thus continually changing position relative to the vibration pickup attached to the bearing.

Figure 6-28
Oscilloscope display of a bearing defect



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This variation in the amplitude may appear as regular amplitude modulation of a sinusoidal oscillation or may show somewhat random amplitude variations.

Measurement Procedures

On machines that have no previous history of Spike Energy measurements, you should use the readings for tracking, or trending, changes in bearing condition. A correlation between these Spike Energy readings and bearing failure must be established by actual experience. Once you have established this correlation, however, you have the basis for using Spike Energy measurements to determine the actual bearing condition.

In any case, it is not recommended that you shut down a machine solely on the basis of Spike Energy measurements. You should make this judgment after considering at least one other vibration parameter such as acceleration or velocity. The reason is that periodic Spike Energy measurements can vary by large amounts, because of changes in machine operating condition. Steam flow, turbulence, high pressure air flow, and cavitation, as well as bearing defects, create Spike Energy signals. Therefore, changes in these operating parameters can cause changes in the measured Spike Energy signals.

For example, you might be led to believe that a sudden increase in the Spike Energy reading for a bearing is indicating an imminent failure. But if a subsequent acceleration or velocity measurement shows no significant increase in vibration, you should consider that the sudden rise in Spike Energy levels may be the result of a change in machine operation (e.g., cavitation) rather than bearing deterioration.

The choice of a second criterion for judging machine condition depends on the function of the machine. A very high speed machine is more readily affected by vibrational forces, therefore an acceleration measurement would be a good indicator. For low speed machines, where relative movement affects operation, displacement would be the preferred backup measurement to Spike Energy. Table 6-2 summarizes the relative vibration amplitudes as a function of the measurement parameter and machine condition. Notice that g-SE readings are HIGH for four out of five machine conditions, whereas only two of these conditions actually justify shutdown.

Table 6-2

	VIBRATION LEVELS			
	Disp	Vel	Accel	SE
MACHINE CONDITION				
Machine OK	OK	OK	OK	OK
Watch Machine Bearing Defects OK to Run	OK	OK	OK	HIGH
Watch Machine Bearing Defects OK to Run	OK	OK	HIGH	HIGH
Machine Problems (Analyze/Shutdown)	OK	HIGH	HIGH	HIGH
Machine Problems (Analyze/Shutdown)	HIGH	HIGH	HIGH	HIGH

9.4 HIGH FREQUENCY ENVELOPING AND DEMODULATION TECHNIQUES

Various vendors provide different high frequency spectrum enveloping techniques. Not all of them will be covered in this manual at this time. The following is a list of their titles:

a. gSE Spectrum (By IRD Mechanalysis based in Columbus, OH)
b. Acceleration Enveloping (By SKF Condition Monitoring based in San Diego, CA)
c. SEE (Spectral Emitted Envelope) (By SKF Condition Monitoring based in San Diego, CA)
d. Amplitude Demodulated Spectrum (By CSI based in Knoxville, TN)

Following in Section 9.41 will be a discussion of one of these offerings (gSE Spectrum). Section 9.5 includes example of data taken using both gSE Spectra as well as other of the above parameters.

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9.41 IRD FAST TRACK gSE SPECTRUM:

IRD FAST TRACK.

gSE SPECTRUM.

BY STUART COURTNEY.
MANAGER TECHNICAL SERVICES.
IRD MECHANALYSIS (UK)LTD.

INTRODUCTION TO gSE SPECTRUM

Industrial equipment downtime is often caused by bearing and gear failures. Therefore, it is important that these types of failures be considered in any machine condition monitoring or predictive maintenance programme. The signals caused by bearing and gear faults are often low in amplitude and jumbled together, this makes them extremely difficult to separate out of a spectrum for detailed analysis. Spike Energy™ Spectrum (gSE Spectrum), an already proven technique incorporated in IRD portable instruments. It is used to assess rolling element bearing and gear condition and is the key to analysing these low amplitude signals.

The overall gSE amplitude is obtained by a unique filtering process.It measures the **INTENSITY** of energy of the minute impacts that occur when an antifriction bearing rotates. These impacts are measured in the ultrasonic frequency range i.e. above 5 KHz (300,000 CPM), the resultant NUMBER is a Figure of Merit that can be used as an assessment of a bearing's condition.

Spike Energy Spectrum is used to provide more detailed information about bearing and gear defects than the overall gSE indication.

In the normal narrowband spectrum there are many components that can be considered as quite normal and acceptable. The primary purpose of gSE Spectrum is to detect which element(s) of a bearing or gearbox may be deteriorating, the vibration energy of these faults may be hidden in the normal satisfactory spectrum.

Spike Energy Spectrum is a reliable measurement used for machinery diagnostics. It is also effective on low speed machinery such as paper rolls and on machines where there is a high degree of background noise such as may be experienced in reciprocating machines or cavitating pumps.

DISCUSSION

The method of signal processing used for gSE Spectrum provides an enveloping and demodulation technique which can identify periodic impacts. These key signs of a deteriorating rolling element bearing are independent of typical vibration signals. This works in exactly the same way that an audio signal is extracted from the carrier frequency in AM radio. Another advantage of the gSE Spectrum technique is that it can be used to distinguish between the different faults associated with the individual components.

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Before we can discuss the method of analysis used for gSE Spectrum it is important to remember that the forces produced by the normal low frequency defects are generally in the form of a steady push and are sinusoidal in their nature.

When a defect develops on an internal surface of a rolling element bearing, bursts of energy are generated each time contact with the defect is made. This information will be present in the measured vibration signals and the bursts will be repetitive at a rate determined by the type of defect, the bearing geometry and the speed of the machine. The frequencies of these bursts are very often the same (or nearly the same) frequencies as that of the steady push frequencies generated by the unbalance, misalignment etc.

These repetition rates are known as the bearing defect frequencies. There are 4 main defects associated with the rolling element bearing :

(1) Ball passing frequency - outer race (BPFO)
(2) Ball passing frequency - inner race (BPFI)
(3) Ball spin frequency (BSF) for a fault on the ball itself.
(4) Fundamental train frequency (FTF) for a cage fault

These characteristic frequencies in CPM can be calculated as follows:

$$BPFO = \frac{1}{2}.RPM.N.\left(1-\frac{Bd}{Pd}.\cos\phi\right)$$
$$BPFI = \frac{1}{2}.RPM.N.\left(1+\frac{Bd}{Pd}.\cos\phi\right)$$
$$BSF = \frac{1}{2}.RPM.\frac{Pd}{Bd}.\left(1-\left(\frac{Bd}{Pd}.\cos\phi\right)^2\right)$$
$$FTF = \frac{1}{2}.RPM.\left(1.\frac{Bd}{Pd}.\cos\phi\right)$$

Where RPM = rotating frequency, CPM
N = number of rolling elements per race
Bd = rolling element or ball diameter, inches(mm)
Pd = rolling pitch diameter, inches (mm)
ϕ = contact angle, degrees

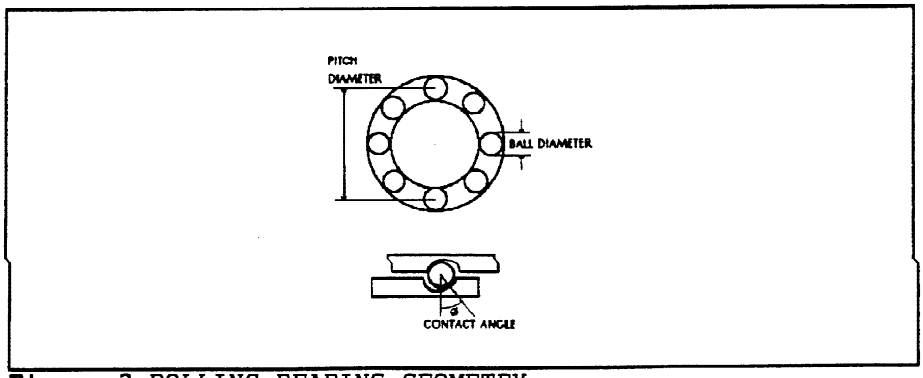

Figure 2 ROLLING BEARING GEOMETRY.

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All of these detected frequencies can be similar to many other frequencies found in the rotating elements of a machine. As these other defects can generate much higher signal levels, it becomes a very difficult task to extract meaningful bearing information. Bursts generated by bearing defects tend to excite natural frequencies of the machine structure (i.e. bearing block) or the transducer. Hence, a signal resulting from these bursts (or spikes) appears as periodic bursts of high frequency energy at intervals determined by the actual bearing defect. In effect, the bearing defect frequency modulates the resonant frequency of the system and the result is a very low level high frequency component. Thus, the spectral waveform may show only a low level amplitude at the resonance frequency and it is typically lost in the background noise.

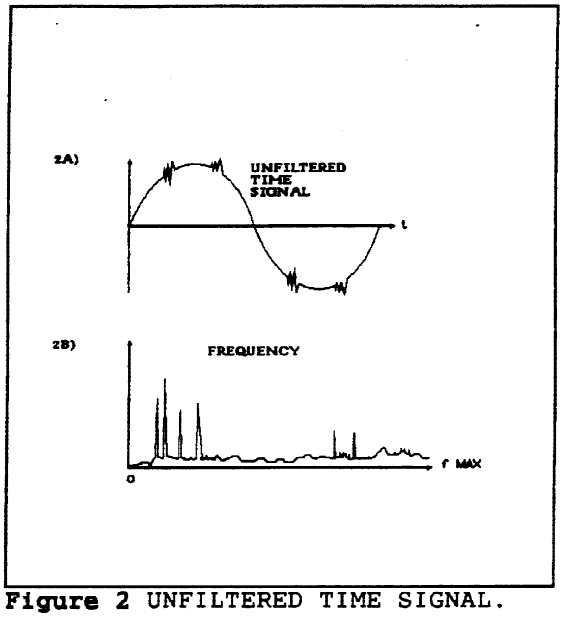

Figure 2 UNFILTERED TIME SIGNAL.

Getting to the details of the defect frequencies requires a three-step process. First, the time domain signal is filtered via a band-pass filter with a selectable low frequency corner. It can be seen in 3A that the low frequency sinusoidal type components are removed from the Time Waveform and if a spectrum analysis was carried out on this information it would simply remove the low frequency components as shown in 3B.

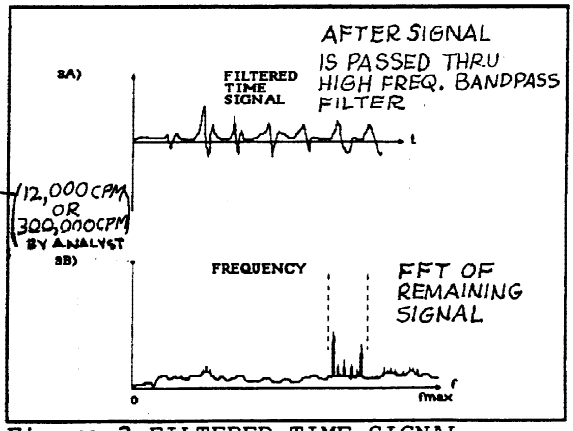

Figure 3 FILTERED TIME SIGNAL.

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In order to identify the low frequency repetition type signals that cause the high frequency excitation of the machine components, the signal is routed through a peak-to-peak detector to extract the repetition rate relating to the particular bearing defect.

Lastly, the spectrum of this demodulated signal is developed to show the particular bearing defect frequency and its harmonics.

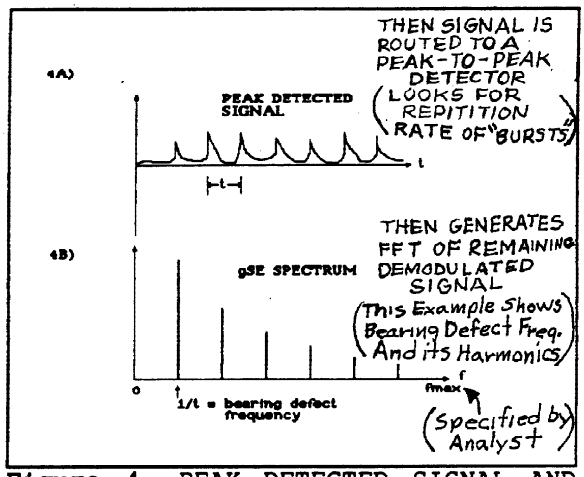

Figure 4. PEAK DETECTED SIGNAL AND SPECTRUM OF gSE

The Spectrum in Figure 4a. is a gSE Spectrum taken with the **IRD FAST TRACK** DATA COLLECTOR; it can be seen that the impact energy caused by a the Ball Pass Frequency Inner of an SKF type 206 bearing in very early stages of failure. Harmonics are present and there is no sign of the low frequency steady push vibration that would be present in the narrow band spectrum. It is interesting to note that the frequency corresponds to the 8 x fundamental caused by the vane pass frequency, this would mask the bearing frequencies in the narrow band spectrum. This method of signal processing obviously does not replace the narrow band spectrum method of analysis. gSE Spectrum is an extra tool in the hands of the vibration analyst that allows assessment of the condition of a machines bearings and gear components.

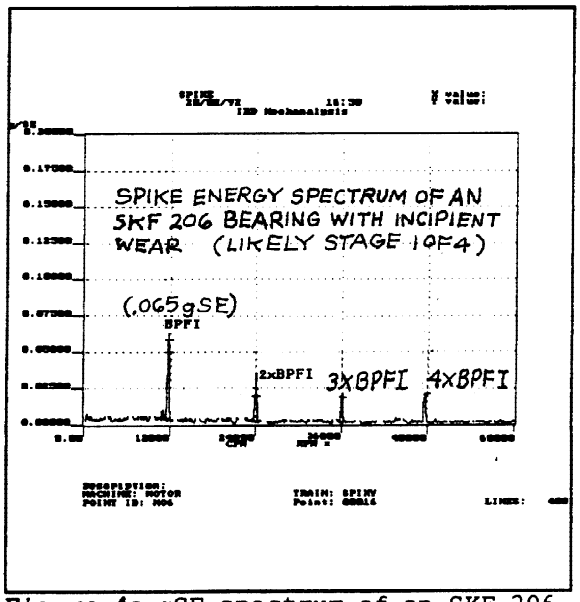

Figure 4a gSE spectrum of an SKF 206.

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FAST TRACK SETUP PARAMETERS.

The uniqueness of the IRD gSE spectrum method of signal processing allows the operator to choose the setup menu that most suits the application.

The gSE spectrum setup menu allows selection of the High pass corner frequency.

The **LOW** corner frequency sets the filter to 200Hz. (12,000 cpm) This setting will allow detection of defects on low speed bearings such as paper rolls.

When the **STANDARD** corner frequency is selected, the filter is set to 5KHz. (300,000 cpm)

The **DECAY TIME CONSTANT** is used to ensure that accuracy of measurement is maintained for all measurements. It is automatically switched depending on the chosen FMAX.

SUMMARY.

In general, the first detectable signs of bearing deterioration are evident in a gradual increase in vibration levels above 2KHz. The faults make contact with other surfaces and structural or transducer resonances are excited and appear as part of the spectrum. These bursts (spikes) cause a modulation frequency that is directly related to the fault causing it. Once the analog signal is passed through a band-pass filter with the appropriate low frequency corner and demodulated by the peak-to-peak detector, the resulting spectral information can be analyzed in detail to determine the specific bearing component deterioration.

As a "normal" rolling bearing will exhibit some vibration, care must be exercised to accurately determine and assess the severity of bearing deterioration. This can be done with personal experience, machine history and by establishing a threshold level for severity assessment. Once a noise background level has been established, the spectral amplitudes can be examined to determine the onset of bearing deterioration and to establish the required corrective action.

Using an **IRD FAST TRACK DATA COLLECTOR** it is a relatively easy task to make these gSE Spectrum measurements to determine early signs of bearing and gear failures. However, it is important to adhere to a few rules. First, all measurements should be made with an accelerometer that is stud mounted. If this is not possible, a 35 Kg magnet mount is (65 lb) acceptable. Second, the IRD Fast Track Data Collector/Analyzer uses selectable high pass filter settings. The selected high pass filter frequency depends on the overall noisiness of the vibration signal. As a rule of thumb, the lowest cutoff frequency that will extract the signal envelope is the best, as it will not be as demanding on the selected transducers performance.

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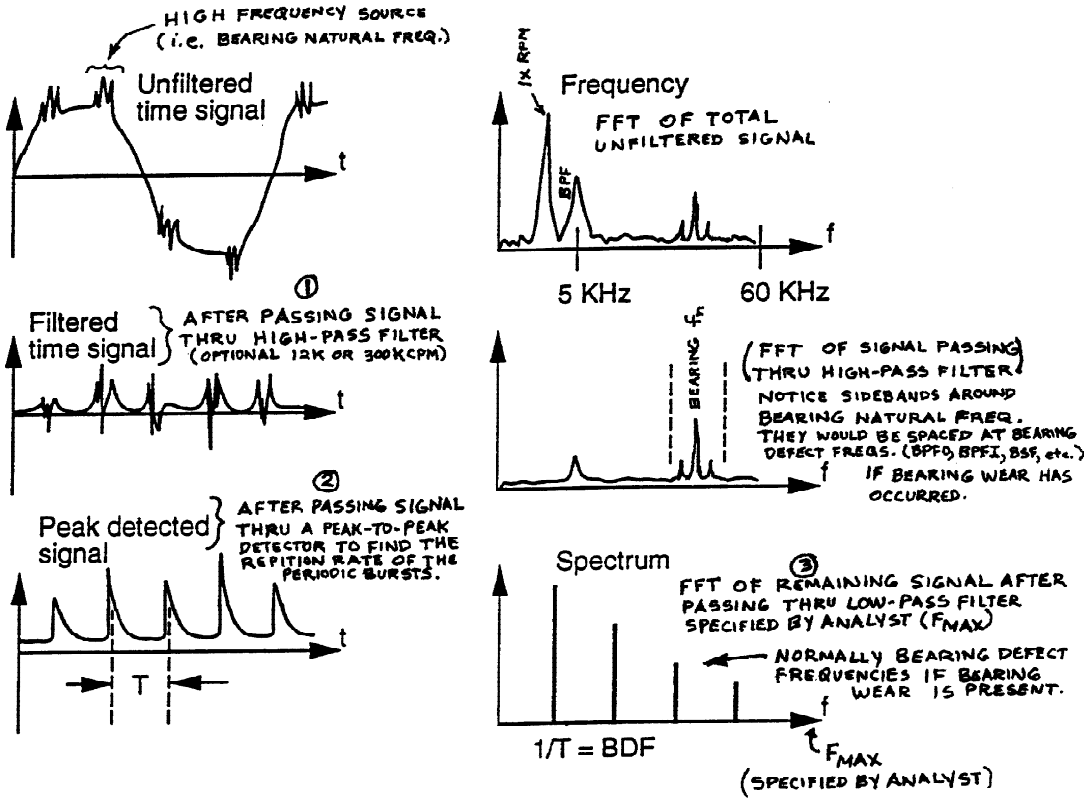

ILLUSTRATION B
SPIKE ENERGY SPECTRUM 3-STEP PROCESS
(PERFORMED ON THE TIME WAVEFORM)

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9.5 CASE STUDIES

9.51 TESTIMONIAL ON THE VALUE OF SPIKE ENERGY SPECTRA

August 21, 1992

Technical Associates of Charlotte, Inc.
347 North Caswell Road
Charlotte, NC 28204

Attention: Jim Berry

Dear Jim,

Enclosed are the velocity and spike energy spectra you had asked for. There are a total of four different machines with a velocity and spike energy spectrum for each one. Each machine had a damaged bearing as indicated by the spectra, but it should be noted that the defects were no more than a stage two bearing failure. I personally feel that if the bearing were allowed to run longer, the velocity spectrum would have indicated the bearing flaw to a greater degree. I am pleased, however, with the spike energy spectrum as an early bearing flaw indicator. I have received and installed the latest Entek software update version 3.27, which supports the spike energy spectrum in the route mode, and hope to soon have a database of gSE spectra to help improve the predictive maintenance program at our mill.

If I can be of further assistance to you, please let me know.

Sincerely,

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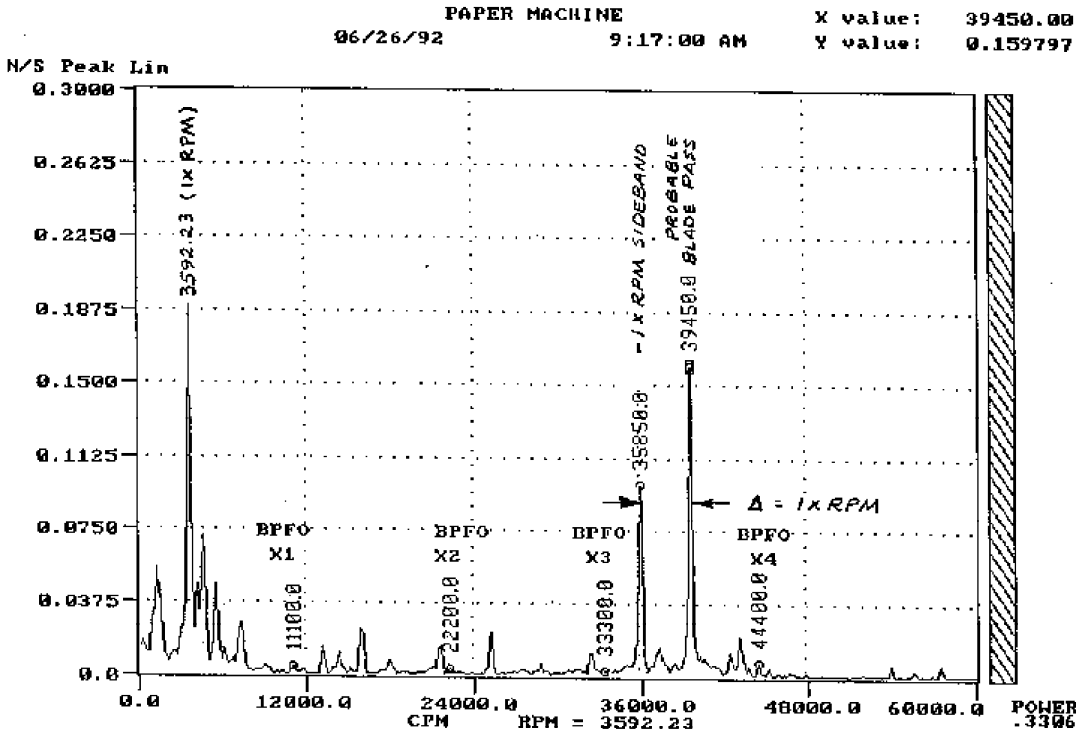

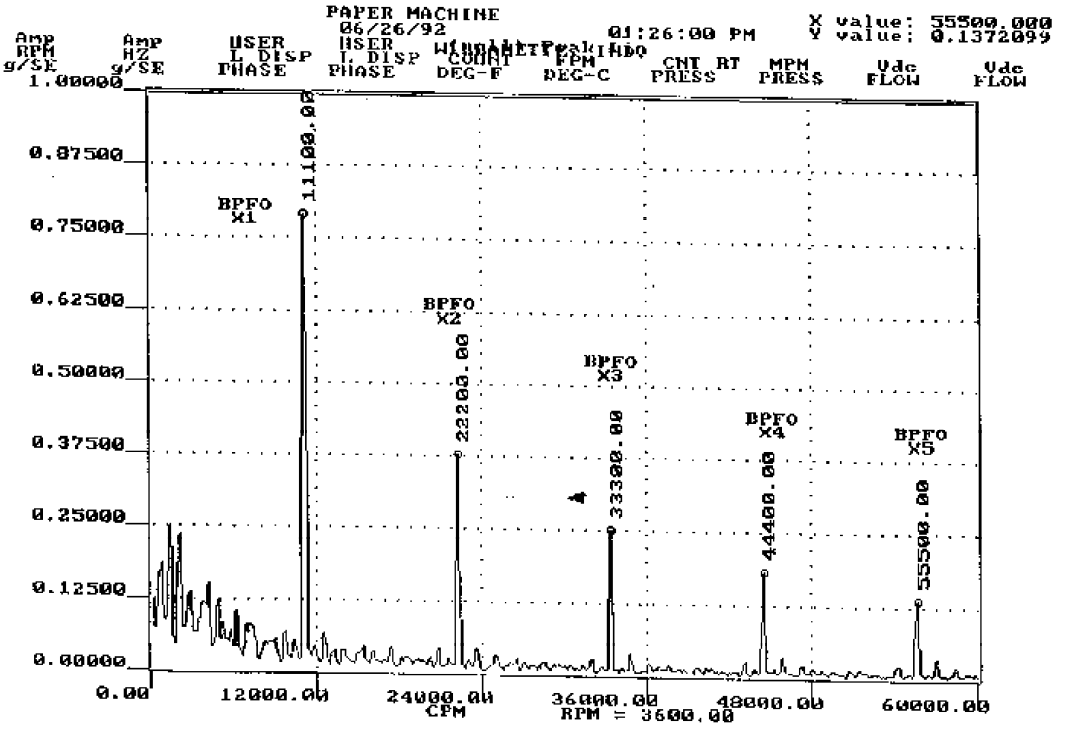

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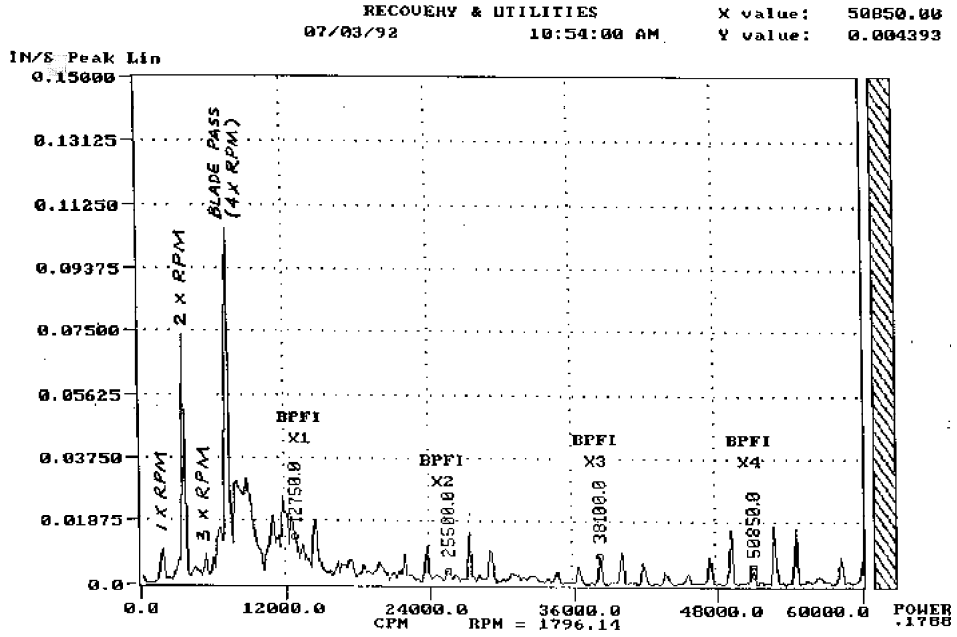

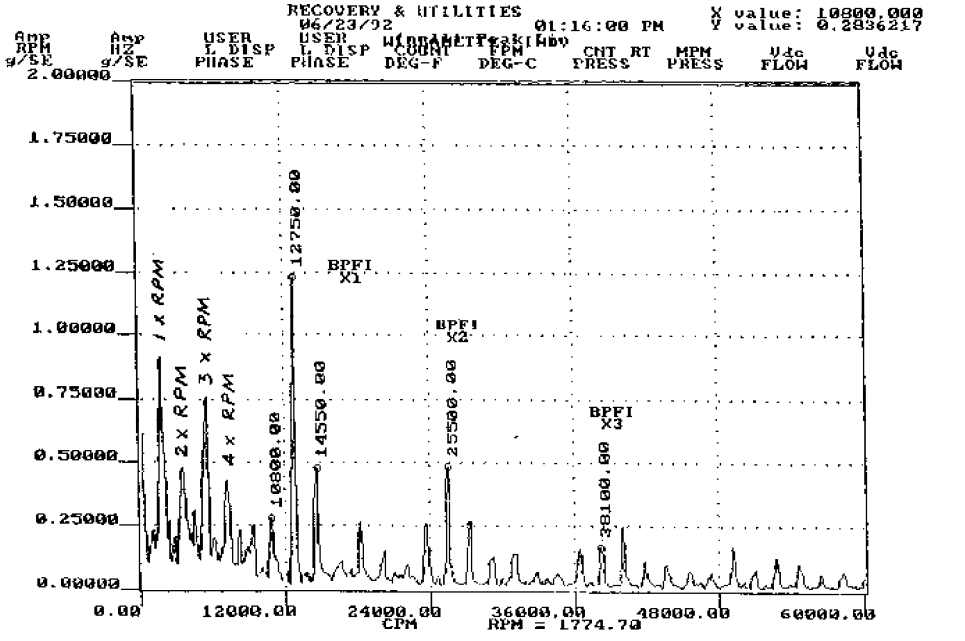

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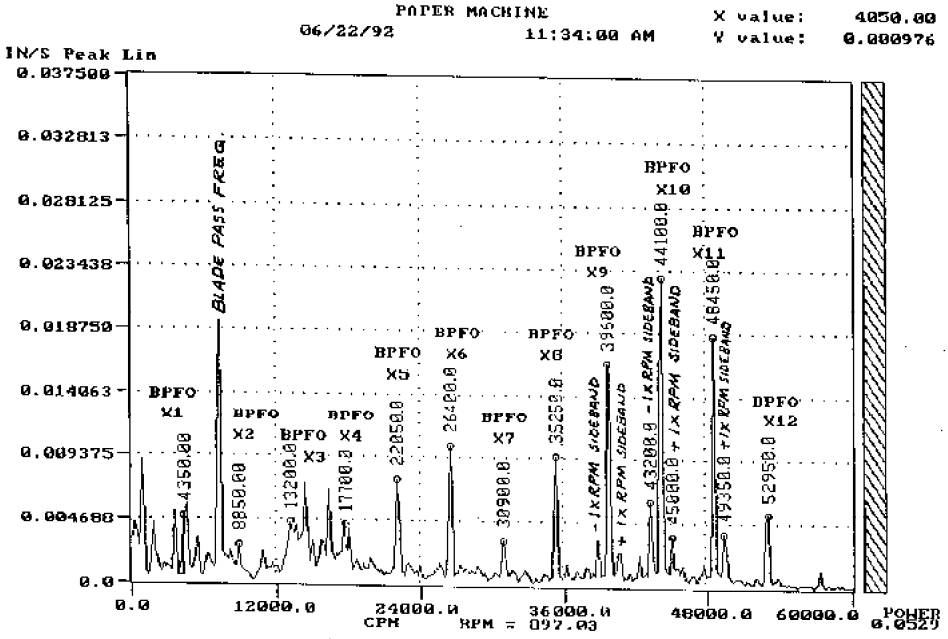
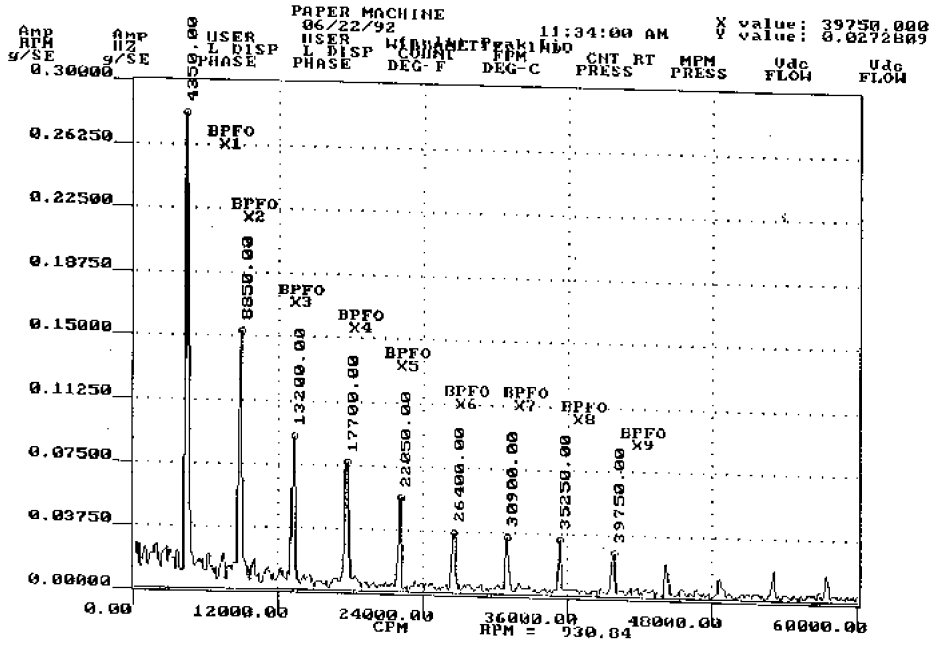


Description: 341.6153 WORTH FARM#3
MACHINE: BROKE STORE PMP TRAIN: STOCK PREP 3 Vertical
POINT ID: PMP DE V WF3 VA Point: 00365



Description: 341.6153 WORTH FRDM#3
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POINT ID: PMP DE V WF3 VA Point: 00369

LINES: 400

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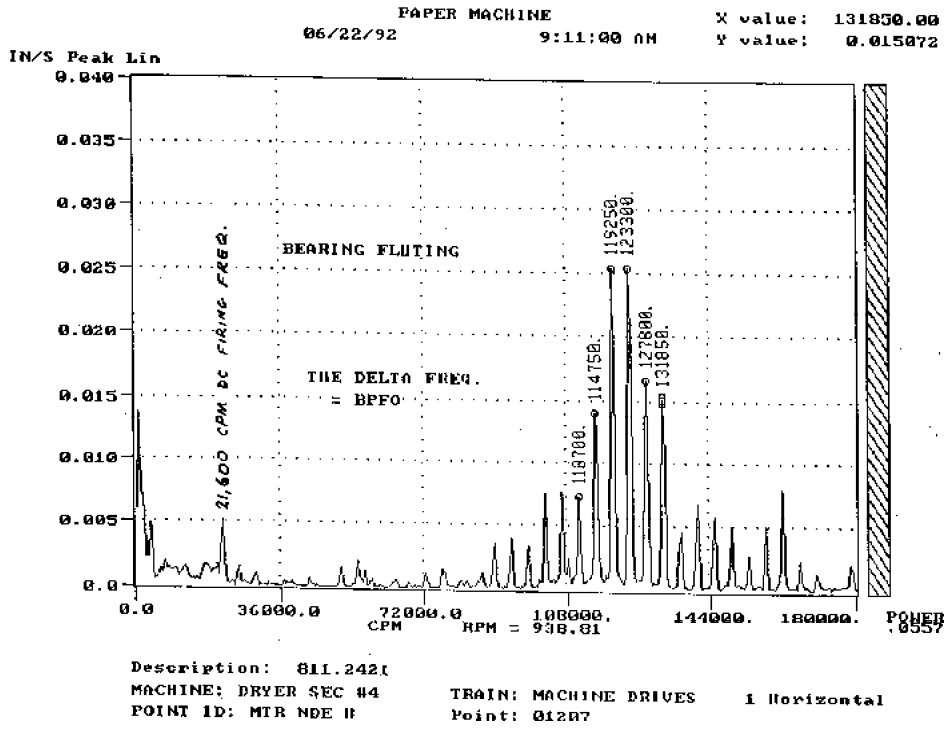

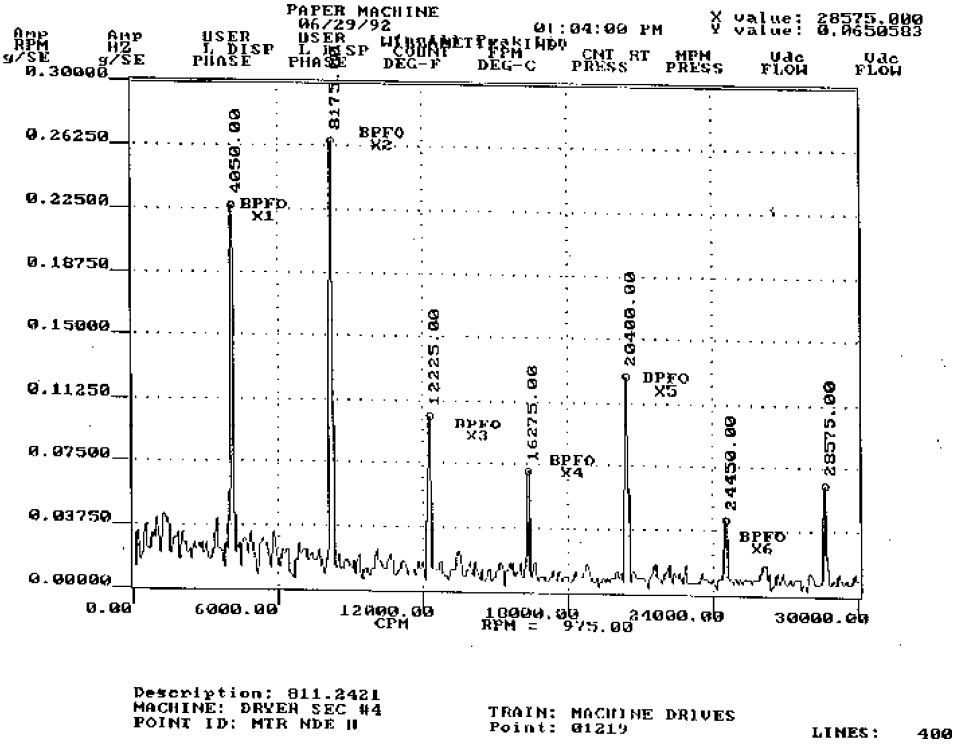

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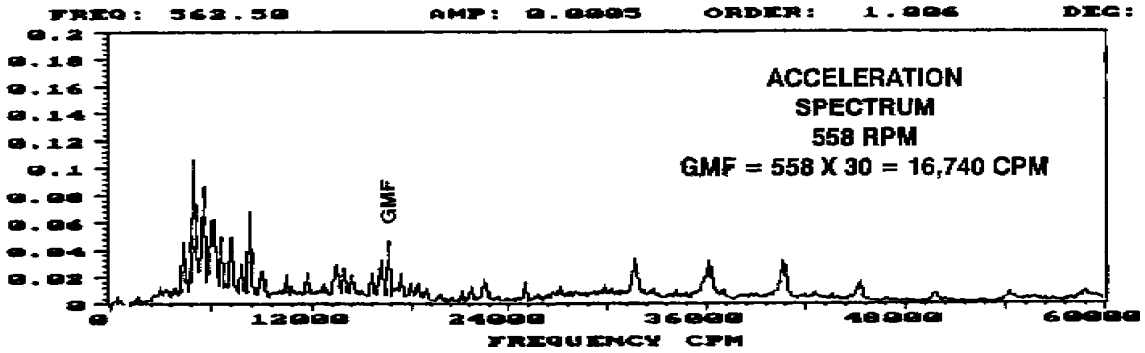
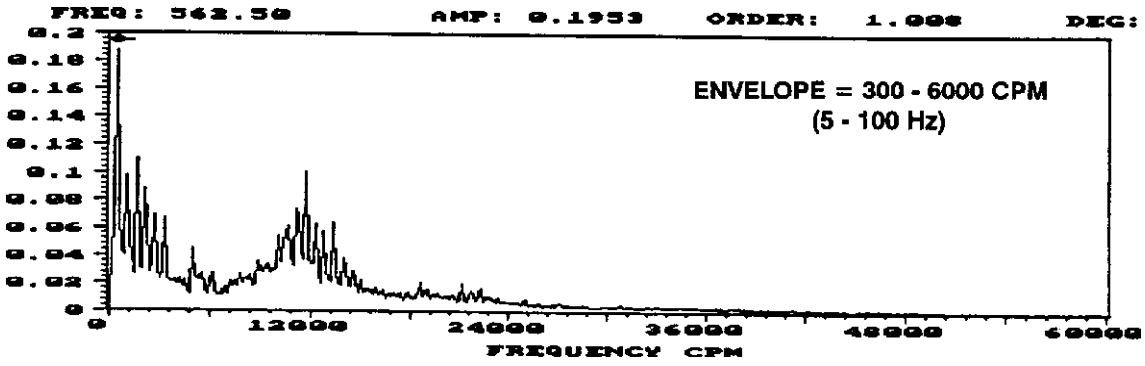
9.52 ENVELOPING (CASE STUDY #2):

Acceleration Enveloping is an extremely useful technique for detecting impulse producing faults such as bearing and gear mesh defects. It is very important to select the proper bandpass filter to facilitate proper implementation of the technique. T*he suggested lower cutoff of the bandpass filter is at least 10X running speed*. This case show the importance of the proper filter selection for Acceleration Enveloping.

Case 2 monitors a Paper Machine Dryer Pinion with 30 teeth rotating at 558 RPM. As can be seen from the 6400 line acceleration spectrum below, a clear fault cannot be observed in the vibration data. The main observations which can be made are (1) harmonics spaced at running speed from about 2500 CPM to about 21,600 CPM; (2) other blunt peaks in the upper range of the spectrum.



Acceleration Enveloping was developed to separate machinery faults which generate vibration impulses such as bearing faults and gear meshing problems, from large carrier signals which are often associated with normal machinery operation. Below is a spectrum showing an Enveloped Acceleration Signal using the 300 CPM to 6000 CPM envelope. Normally the envelope is chosen such that the low end cut off is at least 10X greater than the running speed. At a running speed of 558 CPM, this would mean the envelope should have a lower range of at least 5580 CPM. In this case, little enhancement of the signal can be seen in the vibration signal if a fault were present.



The following figure shows the enveloped signal using the 3000 CPM to 60,000 CPM bandpass filter range. The lower end of the bandpass filter is about 5X running speed. Again this fails the criteria for the lower range of the bandpass filter to be above 10X rotational speed.

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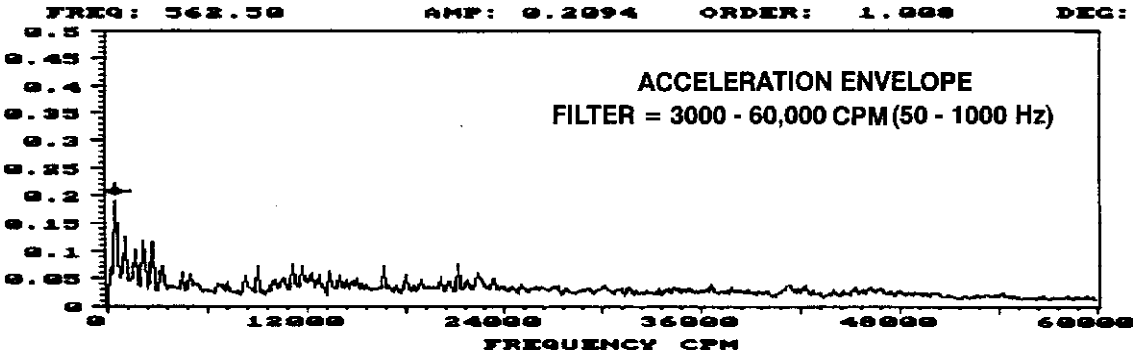
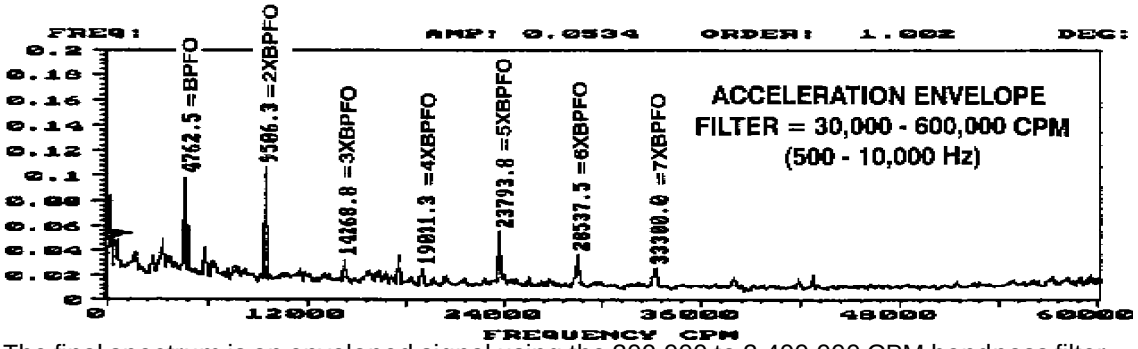
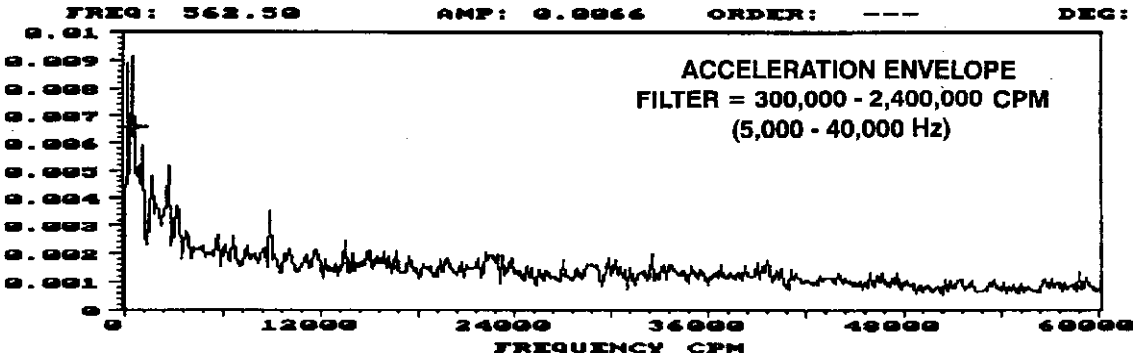


Clearly, the following enveloping filter range, 30,000 CPM to 600,000 CPM, is the proper range forthis signal, as only peaks relating to an outer race defect are evident. This is also the first filter range with a lower cutoff value greater than 10X the nominal running speed. *The signal is much clearer than previous and analysis shows harmonics of 4762.5 CPM relating to an outer race detect for an SKF 22228CCK bearing*. The low amplitude enveloping signals at the calculated outer race defect frequency indicate early detection of an outer race defect.



The final spectrum is an enveloped signal using the 300,000 to 2,400,000 CPM bandpass filter. The lower range of the bandpass filter is over 500X running speed and clearly unacceptable for an enveloping bandpass filter range.



The bearing was removed from service during a down week for the paper machine and inspection showed some water damage and surface cracking in the outer race. During normal operation this bearing would have been left in service until trend levels had increased, but as scheduled down time is opportune, it was removed.

A Division of SK

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CHAPTER 10

INTRODUCTION TO VIBRATION ISOLATION MECHANISMS

Often, vibration analysts and those not associated with vibration technology will erroneously use the terms “vibration isolation” and “vibration damping” interchangeably. In reality, these are two distinctly different mechanisms having two different purposes and results. First, let us take a look at the definition of vibration isolation and point out how it differs from vibration damping:

- **Vibration Isolation -** The major, overriding purpose of vibration isolation is to prevent the unwanted transmission of vibration from one structure or machine to another by “mismatching” the vibration forcing frequencies of the machine or structure from the natural frequency of the isolator mechanism (see Figure 1). One of the key points is that placing isolators between a machine and supporting floor or other structure will most often not reduce the vibration within the machine (in fact, it will increase vibration levels in most machines when they are placed on isolators, typically by about 20% to 40%). Instead, it will simply reduce the transmission of vibration from this machine to surrounding structures; or, on the other hand, will reduce the incoming vibration from other sources into the machine or structure which is to be isolated. It should be pointed out that even though vibration often increases on a machine when it is placed on isolators, no more damage is being done since no additional forces have been applied to the machine - it now simply has less dynamic resistance under the same loads to which it was formerly being subjected (recall that vibration amplitude is proportional to “dynamic load divided by dynamic resistance”).

  Damping is still somewhat related to isolation due to the fact that the higher the damping, the lower will be the vibration response when the forcing frequencies within the machine to be isolated happen to coincide with the isolator natural frequency, either as the machine comes up to speed or coasts to a stop. However, as the damping of an isolator increases, its effectiveness will diminish (see Figure 2). Damping itself is “the dissipation of vibratory energy with motion or with time, converting this energy from mechanical motions to other types of energy, chiefly heat”. Damping treatments are usually only effective when the structure on which they are placed is vibrating at, or very near, resonance.

- **Why are Isolators Needed? -** One of our client plants in the northeast USA has a total of ten (10) large Roots blowers (lobe-type). Normally, he only needs to operate five of the ten machines. In so doing, he found that he began having problems, not necessarily with the machines which were running, but instead, with the machines which were sitting idle most of the time. Upon investigation, he found that the rolling element bearings in the idle machines were being subjected to “false brinelling”, which leaves its “fingerprints” behind as a series of indentations particularly in the load zone of the outer race, with the spacing of the indentations equal to the spacing between each of the rolling elements. If this false brinelling is allowed to continue, these indentations can become quite pronounced. Therefore, when it was necessary to operate these machines, the bearings themselves already could have significant damage on their outer races, which could easily transfer into the rolling elements, and from the rolling elements into the inner race of the bearing bringing about very short-lived life of these bearings. To prevent this from reoccurring required him to install vibration isolation systems on each of the ten Roots blowers in this case. Since doing so, this has virtually eliminated the false brinelling problems and premature bearing failures these machines were suffering before.

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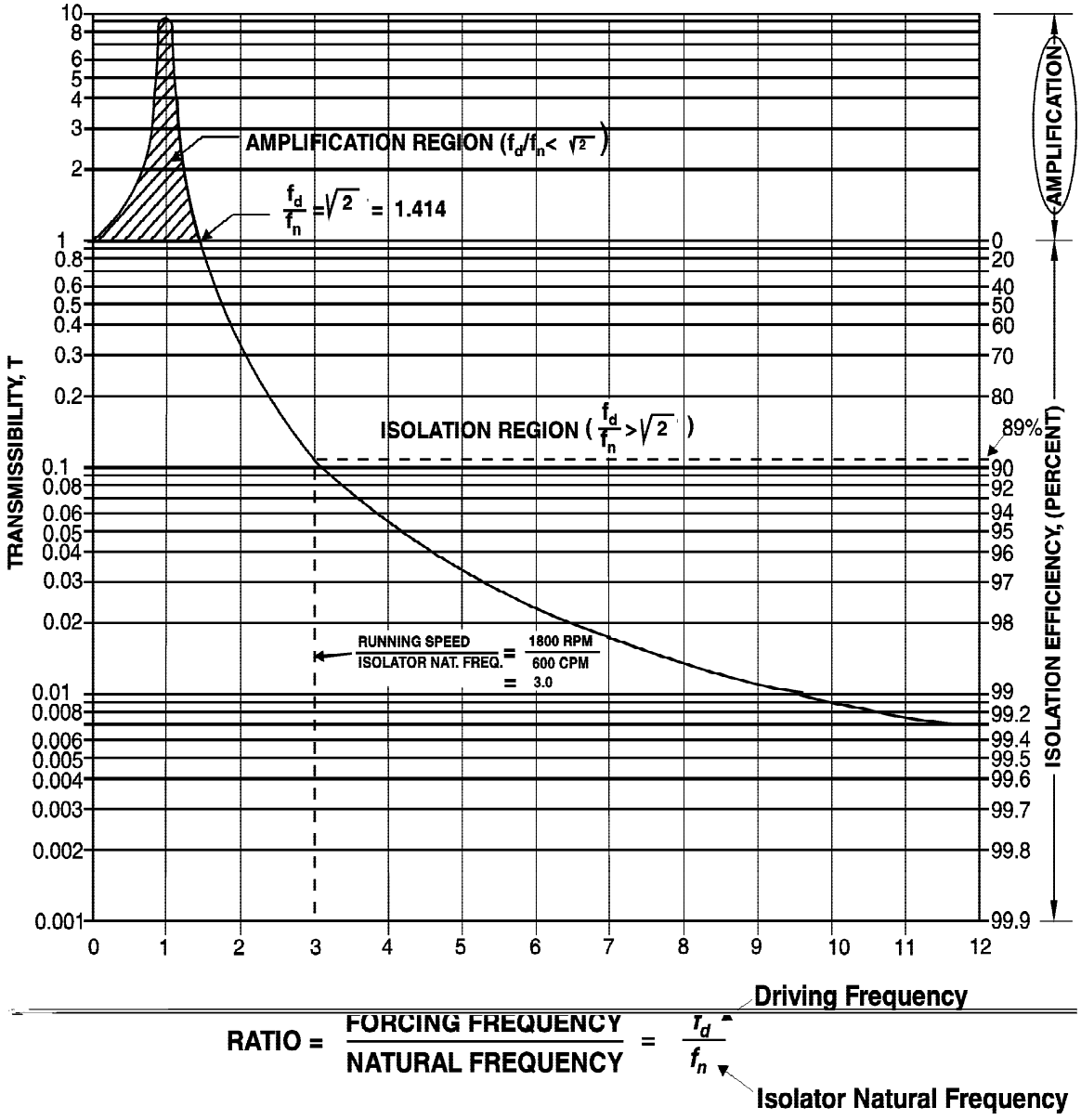
FIGURE 1
THEORETICAL VIBRATION ISOLATION EFFICIENCY &
TRANSMISSIBILITY CURVE (ASSUMED DAMPING FACTOR z = .05)
(Ref. 1)



EXAMPLE: IF OPERATING SPEED = 1800 (30 Hz) FOR A DIRECT - COUPLED FAN AND ISOLATOR NATURAL FREQ. = 600 CPM (10 Hz)

FIND: THEORETICAL ISOLATION
FROM GRAPH ABOVE, ISOLATION = 89% (FOR A 3:1 RATIO)

IF AN ISOLATOR WITH A NATURAL FREQ. OF 300 CPM (5 Hz) COULD BE INSTALLED:
1800 RPM / 300 CPM = 6:1; THUS, ISOLATION = 97.2% (FOR A 6:1 RATIO)

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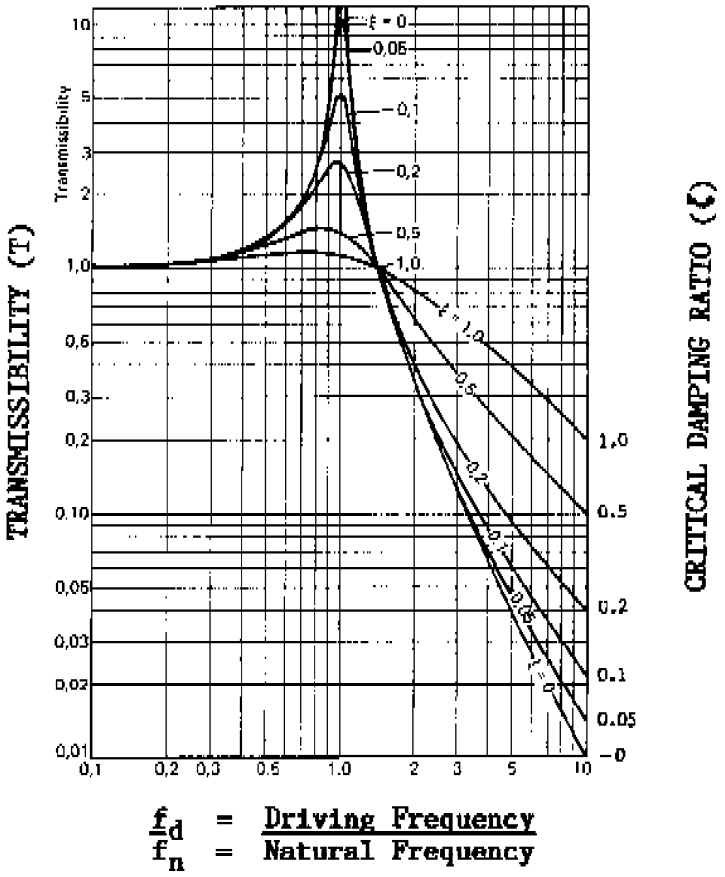
FIGURE 2
THEORETICAL TRANSMISSIBILITY CURVE FOR A RANGE OF DAMPING FACTORS (z) IN A SINGLE DEGREE-OF-FREEDOM SYSTEM (Ref. 2)



$\frac{f_d}{f_n}$ = $\frac{\text{Driving Frequency}}{\text{Natural Frequency}}$

$$\text{TRANSMISSIBILITY} = T = \sqrt{\frac{1 + \left[(2\zeta)\left(\frac{f_d}{f_n}\right)\right]^2}{\left[1 - \left(\frac{f_d}{f_n}\right)^2\right]^2 + \left[(2\zeta)\left(\frac{f_d}{f_n}\right)\right]^2}}$$

where:

T = Transmissibility
f_d = Driving or Forcing Frequency
f_n = Natural Frequency
$\zeta = c/c_c$ = Critical Damping Ratio

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In other cases, it might be necessary to install a highly precision measurement instrument such as an optical comparator or an electron microscope in a laboratory where the floor is subjected to noticeable vibration. This vibration might be generated by process machinery located outside the lab such as reciprocating air compressors, punch presses, air handling systems, etc. In several cases, the author has been involved when railroad trains operating up to one-half mile away induced sufficient vibration into a precision lab to totally interrupt any measurement processes with his instrument while the train was passing. In these cases, precision measurements had to be made to evaluate the vibration severity; and then, sophisticated vibration isolation mechanisms had to be put in place to virtually eliminate this undesirable transmission of vibration from such background sources. An illustration of one of these isolation mechanisms will be given later in this section using Figures 9 and 10 showing how a fairly simple isolation system was used to prevent fairly substantial vibration originating at a two-stage reciprocating compressor from transmitting into a load bearing wall located less than two feet away from the compressor; and from transmitting this vibration into an electron microscope located directly above this compressor installation on the second floor.

- **How Does Vibration Isolation Work? -** Figure 1 helps explain how vibration isolation works (this figure assumes a damping factor, ζ, equal to .05). In Figure 1, the left vertical axis graphs the vibration "Transmissibility" (T) while the right-hand vertical side graphs "Isolation Efficiency Percentage". The horizontal axis pertains the ratio of forcing frequency (f_d) divided by the natural frequency (f_n). In this case, the natural frequency will actually be the natural frequency of the entire isolated system, but will be almost totally influenced and controlled by the natural frequency of the isolation mechanism itself. In the case of forcing frequency, sometimes called driving frequency (f_d), this frequency can be due not only to operating speed, but also to 2X RPM, 3X RPM, blade pass frequency (# blades X RPM), gear mesh frequency (# gear teeth X RPM), etc. The key point here will be to find the lowest forcing frequency of considerable amplitude and to design the entire isolation system around this lowest frequency of significant amplitude. Whereas in most machines this frequency will normally be at operating speed, please be careful if working with a belt-driven system (particularly a multi-belt driven machine) since the "belt RPM" will always be lower than the RPM of either the driver or driven machine components. When the belts begin to wear, or if they are either severely mismatched or operating at or near a belt resonance, they themselves can be the source of considerable vibration which must be taken into account when specifying a vibration isolator. The key point here is that assuming a single degree of freedom system such as that graphed in Figure 1 (one mass and one spring), if the isolator chosen effectively isolates the lowest frequency of significant amplitude, it will do an even more effective job on isolating higher and higher frequencies. For example, consider a 1780 RPM motor driving a 1000 RPM fan using a six V-belts which themselves turn at approximately 300 RPM. Again, if the belts encounter problems, this frequency which would have to be isolated. If this machine happened to operate near precision instruments, the lowest frequency of significant amplitude very likely might not be the 1000 RPM fan speed, but the belt speed of approximately 300 RPM instead. It should be pointed out that the lower the frequency which must be isolated, the more difficult will be the isolator requirements.

- **What is a Good Rule of Thumb for Specifying Proper Vibration Isolators? -** First, please return to Figure 1. Note on the upper left-hand side of the figure the region denoted as the "Amplification Region". Unfortunately, in plants throughout much of the world, vibration "amplifiers" have been installed rather than vibration isolators. That is, both the machine and the surrounding area would be much better off without any so-called "isolators" since they are in reality "amplifiers". That is, if the forcing frequency of either the machine to be isolated, or of the vibrations transmitting through the floor or surrounding structure into this machine happen to line up with the isolator natural frequency ($f_d/f_n = 1$), the vibration can easily be 10 or more times higher than it would have been were no "isolators" installed whatsoever. In fact, no isolation takes place until the lowest forcing frequency (f_d) is at least 1.414X the isolator natural frequency (f_n). Figure 1 helps demonstrate a good rule of thumb for general rotating machinery. In this case, if an

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isolator can be found where the forcing frequency is at least 3X higher than the isolator natural frequency, fairly significant isolation on the order of approximately 89% will be achieved. In other cases, where even more isolation is required, this ratio might have to be increased to as much as 7 or more to achieve about 98% isolation which is required in the case of many precision instruments. Looking at the example given in Figure 1, note here that this was a direct-coupled fan which was transmitting considerable vibration into the surrounding structure. In this case, an isolator with a natural frequency of 600 CPM (10 Hz) was used under the motor/fan system operating at 1800 RPM which resulted in 89% isolation.

- **How Does the Amount of Isolator Damping Affect Isolator Performance? -** Some isolators can be outfitted or combined with damping materials which helps reduce the vibration response when the machine accelerates to operating speed through particularly the isolator natural frequency. In fact, some steel spring systems must be outfitted with snubbers to prevent the motion from becoming excessive if the machine takes some time to pass through this resonance. The problem with adding damping is that the higher the damping, the less will be the resultant isolation. For example, at a ratio of forcing frequency over natural frequency of 10.0, an isolator with a damping ratio of .02 will theoretically have an isolation of 99.0%; while an isolator with a much higher damping ratio of .25 will have an isolation percentage of about 94% (this may not appear to be much of a difference, but if on a precision gage, it may make the difference). On the other hand, if the frequency ratio were only 3.0, this would theoretically drop the isolation from approximately 90% down to only 75% (meaning that 25% of the vibration would pass through with the higher damped isolator compared with only 10% in the case of the lower damped isolator).

- **What Are Some Typical Types of Isolators and How Does Their Performance Compare?** Table I compares some of the more common isolator types, along with their normal range of natural frequencies. Likewise, Table I shows what range of theoretical forcing frequencies each isolator type will isolate, both for approximately 90% as well as for 96% isolation. That is, if one had a 1780 RPM, direct-coupled fan, he would definitely not want to choose either cork or ribbed elastomeric pads for isolators as shown in Table I. Instead, he should choose either steel springs, elastomeric isolators or fiberglass pads. On the other hand, if he were trying to isolate a lower speed, belt-driven fan of approximately 450 RPM, he would likely be forced to use steel springs or pneumatic mounts as shown in the table. Figures 3 through 7 show some of the typical isolator types used today ranging from the rather inexpensive isolators shown in Figures 3 and 4 up to the costly pneumatic spring systems which are supplied with a continuous air supply as shown in Figure 7. The less expensive isolators such as the steel spring types shown in Figure 5 can be used on machines operating at speeds of 400 RPM and above. However, if the speed drops down to 100 or 200 RPM (or if high percentage isolation is required for a precision instrument) the more expensive pneumatic spring isolators which are provided with continuous air supply will likely be required. These particular units have natural frequencies themselves as low as 0.5 Hz (30 CPM). Most of these mechanisms are self-leveling and self-monitoring to keep each of the air springs supplied with just the right amount of air for both leveling and optimum isolating performance. Using these devices, it is possible to make precision measurements with the instruments, even when the plant might be located near a railroad track, or if there are production machines nearby causing much vibration transmission through the floor such as reciprocating machines, punch presses, etc.

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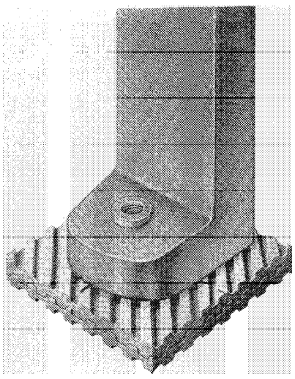
TABLE I.
APPROXIMATE NATURAL FREQUENCY RANGES & PERFORMANCE DATA FOR VARIOUS ISOLATOR TYPES AND MATERIALS

ISOLATOR TYPE	NATURAL FREQ. RANGES (Hz)	FORCING FREQUENCIES THEY WILL ISOLATE			
		≈ 90% ISOLATION		≈ 96% ISOLATION	
		Hz	CPM	Hz	CPM
Cork	20 - 40	66 - 132	3960 - 7920	100 - 200	6000 -12000
Ribbed Elastomeric Pads	15 - 35	50 - 115	3000 - 6900	75 - 175	4500 - 10500
Fiberglass Pads	5 - 15	17 - 50	1020 - 3000	25 - 75	1500 - 10500
Elastomeric Isolators	6 - 12	20 - 40	1200 - 2400	30 - 60	1800 - 3600
Steel Springs	2 - 8	7 - 26	420 - 1560	10 - 40	600 - 2400
Pneumatic Mounts (Non-Cont. Air)	2 - 4	7 - 13	420 - 780	10 - 20	600 -1200
Pneumatic Mounts (Continuous Air)	0.5 - 1.5	1.7 - 5	102 - 300	2.5 - 7.5	150 - 450

NOTE: Natural Frequency Ranges and forcing frequencies for which they will provide 90% and 96% isolation listed above for each Isolator Type are approximate ranges which depend on exact design configuration and load distribution. Consult isolator manufacturer for more precise natural frequency and performance data.

FIGURE 3
RIBBED ELASTOMERIC PAD ISOLATOR
(Ref. 3)



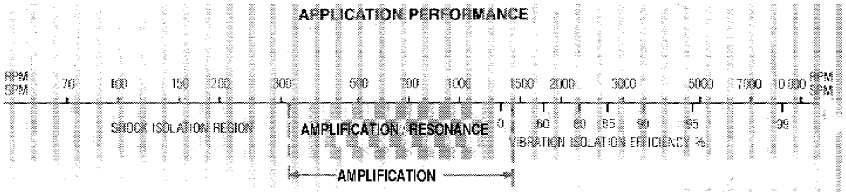
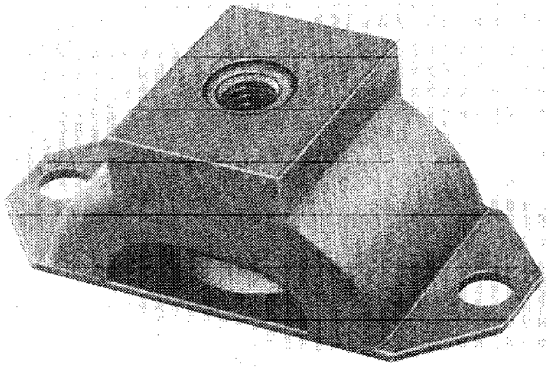
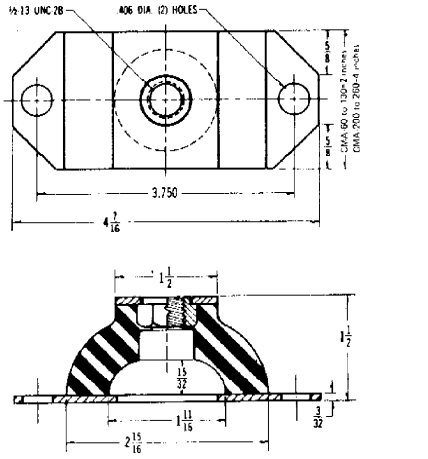


FIGURE 4
ELASTOMERIC ISOLATOR
(Ref. 3)





Installation: The top plate includes a captivated nut tapped 1/2-13 UNC-2B for ease of attaching equipment. Through holes on each side flange accommodates ⅜" tie-down bolts for secure attachment to support structure. (Bolts not supplied.)

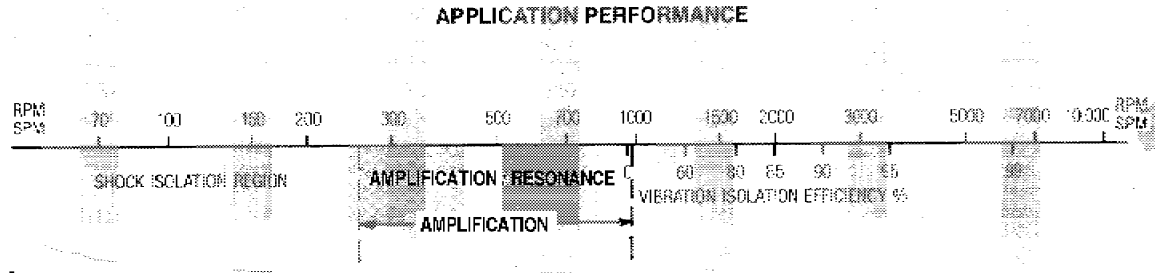


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FIGURE 5
STEEL SPRING ISOLATOR WITH LEVELING BOLT AND ADJUSTABLE SNUBBING
(Ref. 4)

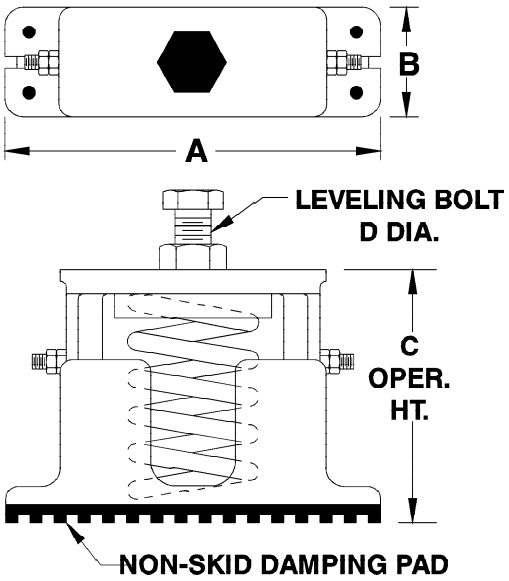

FIGURE 6
PNEUMATIC MOUNTS (NONCONTINUOUS AIR SUPPLY)
(Ref. 3)

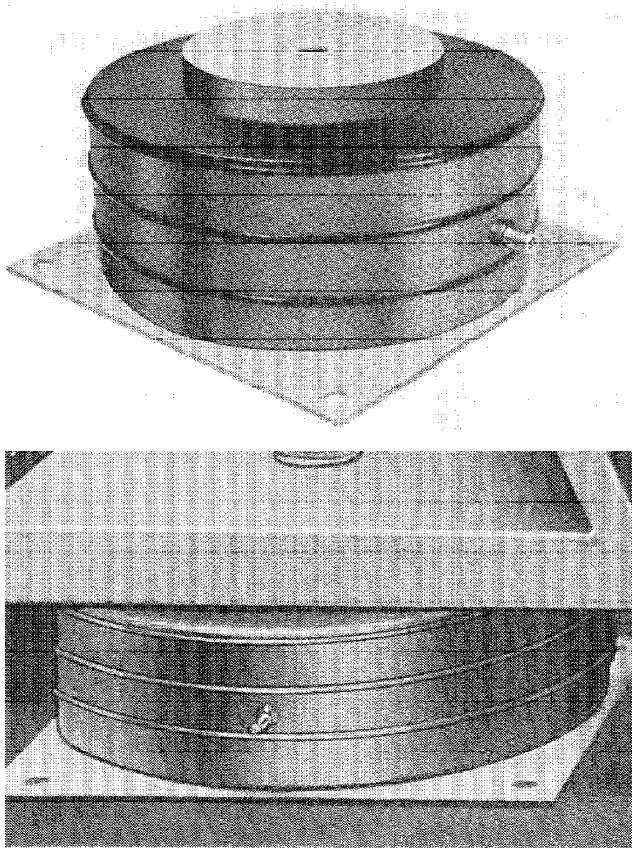

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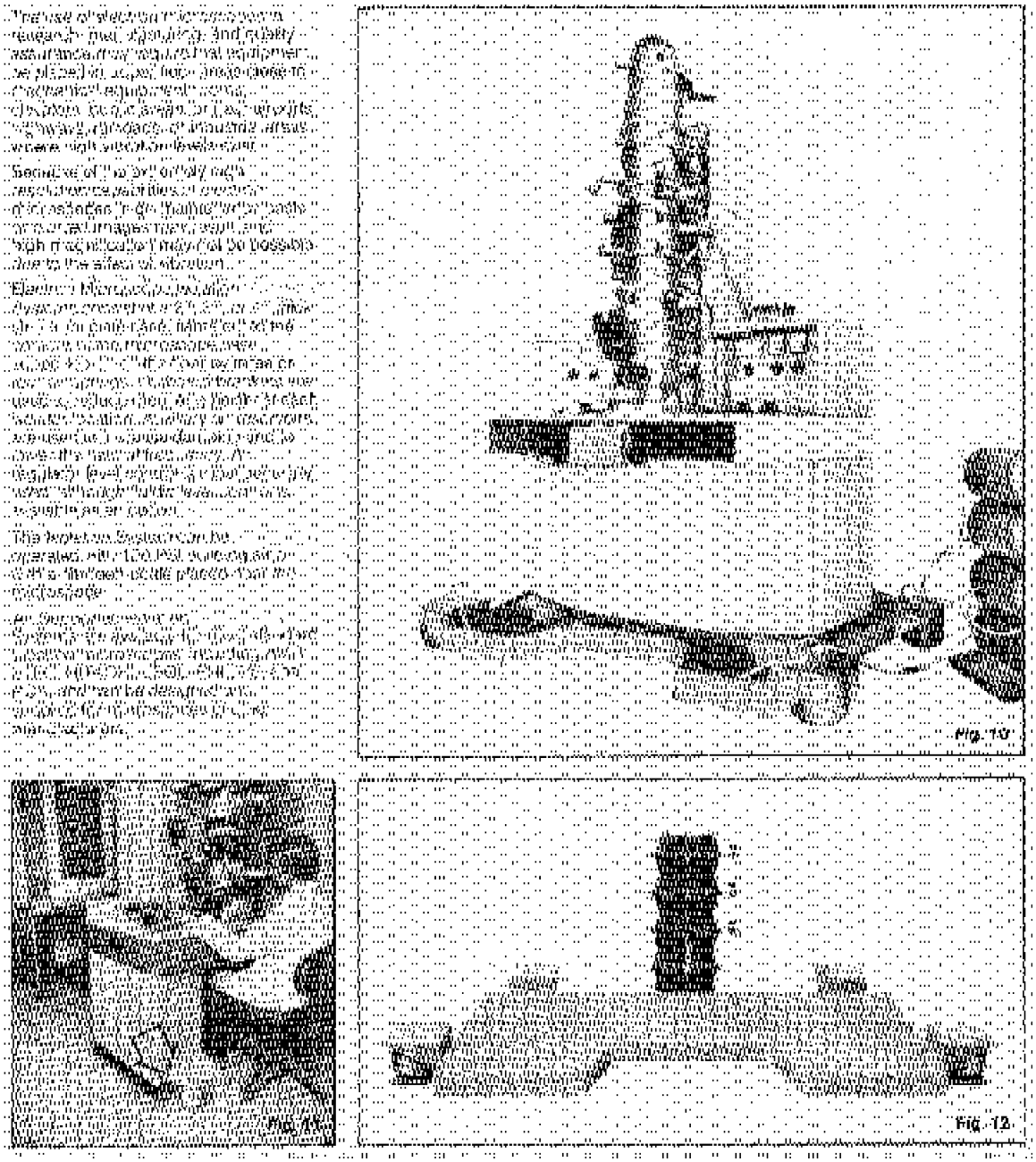
FIGURE 7
PNEUMATIC SPRING WITH CONTINUOUS AIR SUPPLY FOR VERY LOW FREQUENCY ISOLATION & FOR PRECISION MEASUREMENT INSTRUMENTS (Ref. 5)



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- **Real-World Case History - "Provision of an Effective Isolation System to Prevent Transmission of Vibration into an Electron Microscope from a 2-Stage Reciprocating Air Compressor to be Installed on the First Floor Directly Beneath the Microscope Lab"**

  In practical situations, it is not always necessary to expend great funds when needing significant isolation. It only requires some common sense and practical experience. For example, please refer to the isolation system provided for the two-stage reciprocating air compressor pictured in Figures 8 and 9. In this case, this was a used compressor purchased by the client who allowed the compressor to sit out in the weather for over one year before its installation. Then, he decided to locate it on the first floor of a building, only 21" away from a load bearing wall as shown in Figure 9. Right above the compressor on the second floor was an electron microscope. Therefore, several measures were taken to minimize the vibration transmission into the floor and surrounding structure. First, an entire floating isolation frame was fabricated from channel and I-beams as shown in Figure 9. Then, 15" of concrete was poured into the isolator frame creating an inertia block some 72" wide and 126" long. This entire inertia block and isolation frame was then supported by steel spring isolators with leveling bolts and adjustable snubbing like that shown in Figure 5. Also, a number of calculations were made to ensure that the system center of gravity ("SYSTEM CG" shown in Figure 8) would be positioned to be in line with the center of flexure of each of the spring isolators (note "CF" which shows the spring center of flexure in Figure 8). By aligning the system CG and spring center of flexure (CF), this minimized rocking when the compressor was started up and brought up to its 600 RPM operating speed. During this time, it of course passed through the isolator natural frequency which was calculated to be 2.27 Hz (136 CPM). Comparing this with the 600 RPM operating speed, this worked out to a ratio of about 4.4. Referring back to Figure 1, this resulted in an isolation efficiency of approximately 95%. In addition, before installing the compressor isolation frame, the original floor and the top layers of soil were removed. The upper soil layers in the pit were then compacted to a compaction rate of about 90%. Then, layers of polystyrene were inserted before repouring a new concrete floor, but this time with a 1" cork expansion joint installed all the way around the concrete pad as shown in Figure 9. Finally, a layer of .25" thick neoprene vibration damping pads were placed between the concrete floor and each of the isolator spring support stands to help dampen vibrations which did transmit through the isolators. Also, the client was told to maintain at least a 2" clearance between the floor and the floating inertia block to prevent trash and other foreign materials from building up over time, thereby "short circuiting" the isolation mechanism. Finally, flexible connections similar to those shown in Figures 10 and 11 were placed in all piping connected to the compressor system, again to prevent short circuiting of the isolating system. Not only one, but two flexible connections were recommended in each piping circuit spaced at least 2 feet apart, preferably with one oriented in a horizontal run and one in a vertical run. This would effectively break up vibration wave patterns and greatly limit their transmission into adjacent structures.

  After installation of the system shown in Figures 8 and 9, rigorous testing showed that it provided over 98% isolation. This meant that less than 2% of the significant reciprocating forces made their way into the surrounding floor and building structures. Therefore, it was still possible to successfully use the electron microscope on the second floor despite having a used two-stage reciprocating compressor operating directly beneath it, and only 21" away from the load bearing wall passing through these adjoining structures.

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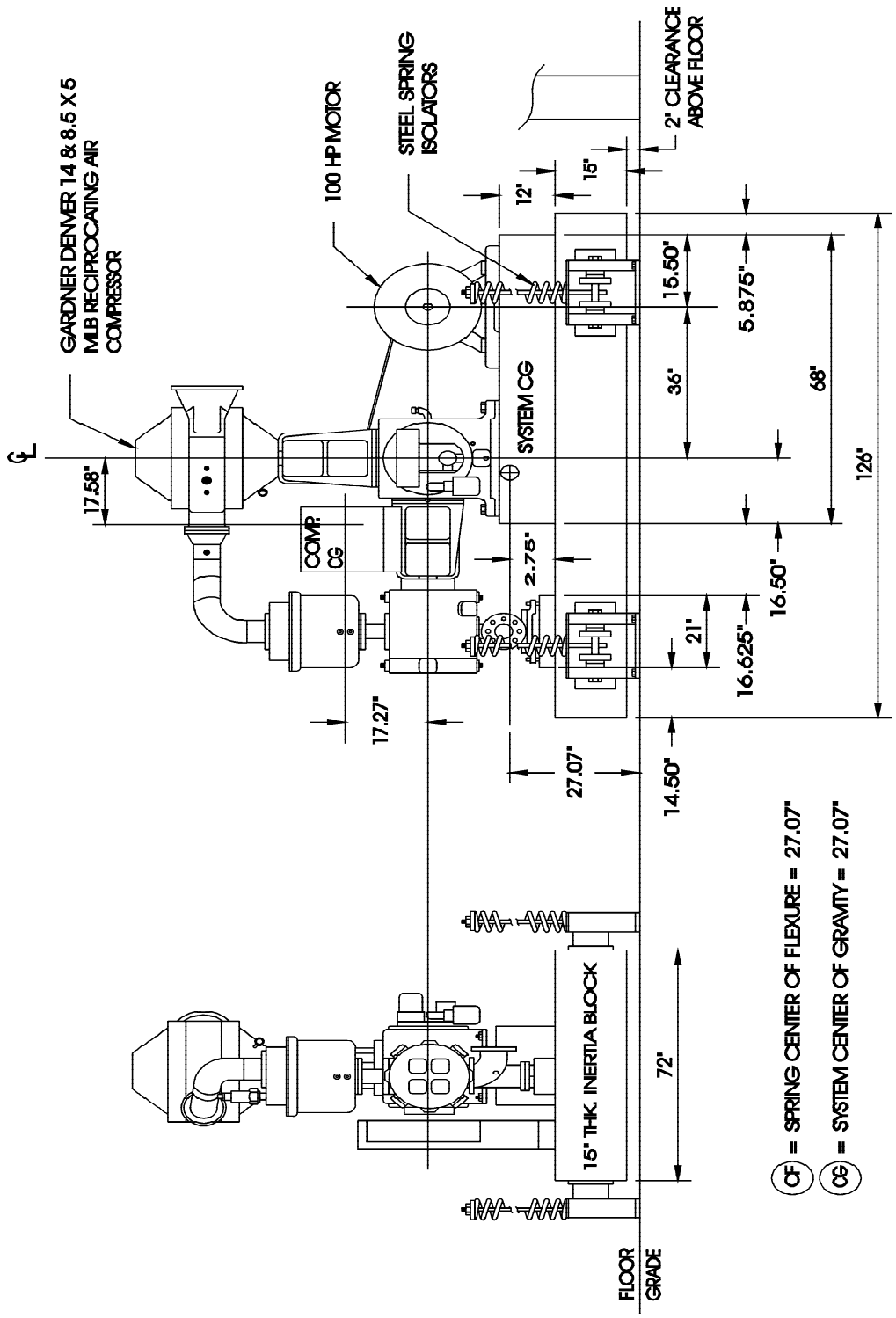

FIGURE 8
COMPRESSOR AND MOTOR ON PROPOSED VIBRATION ISOLATION SUPPORT SYSTEM
(Ref. 6)

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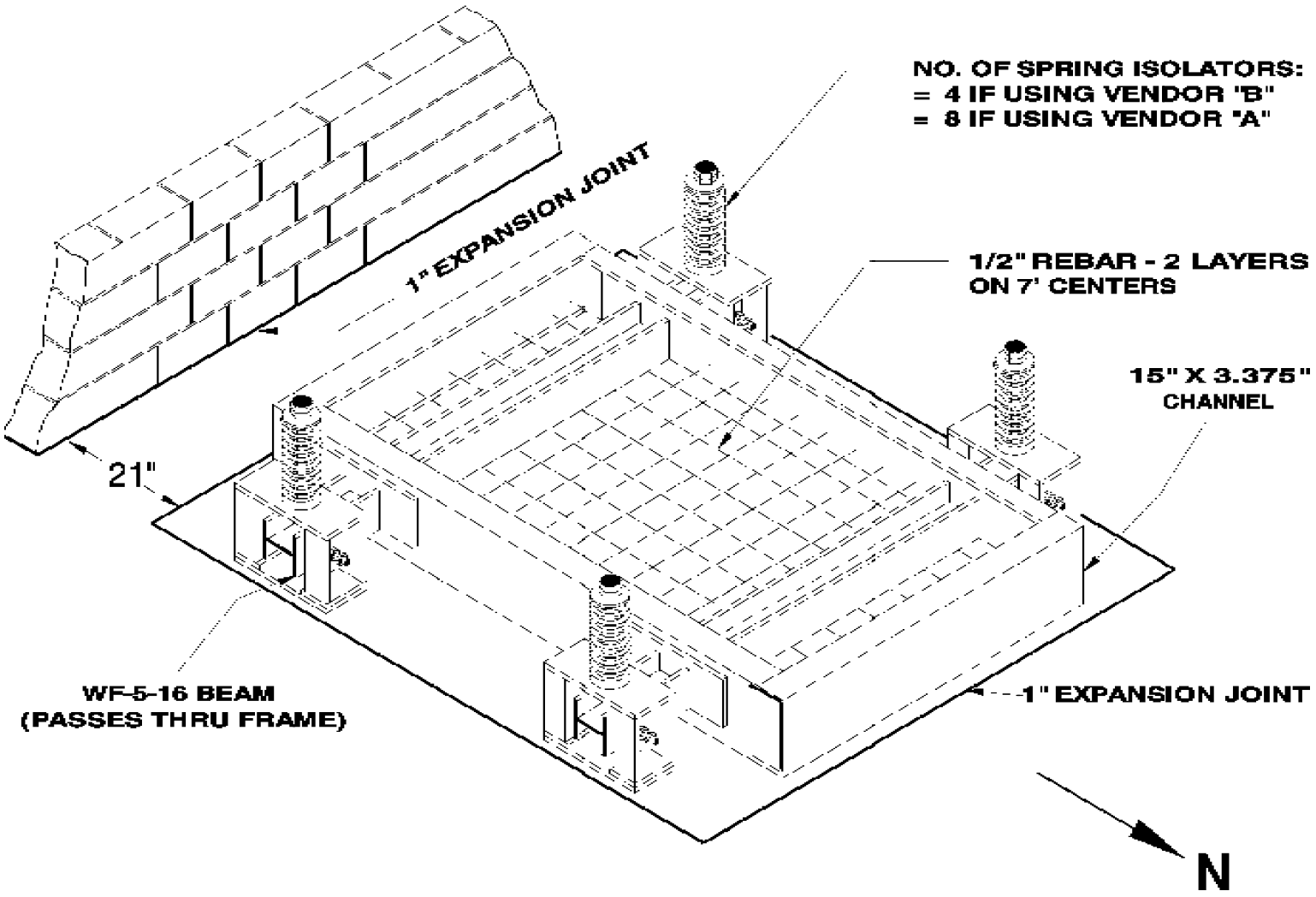
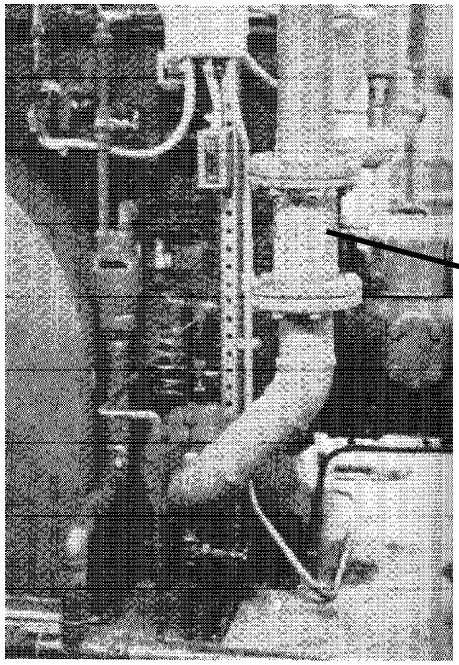


FIGURE 9
PROPOSED VIBRATION ISOLATION FRAME

FIGURE 10
VIBRATION ISOLATION SYSTEM FOR A COMPRESSOR AND ATTACHED PIPING (Ref. 7)



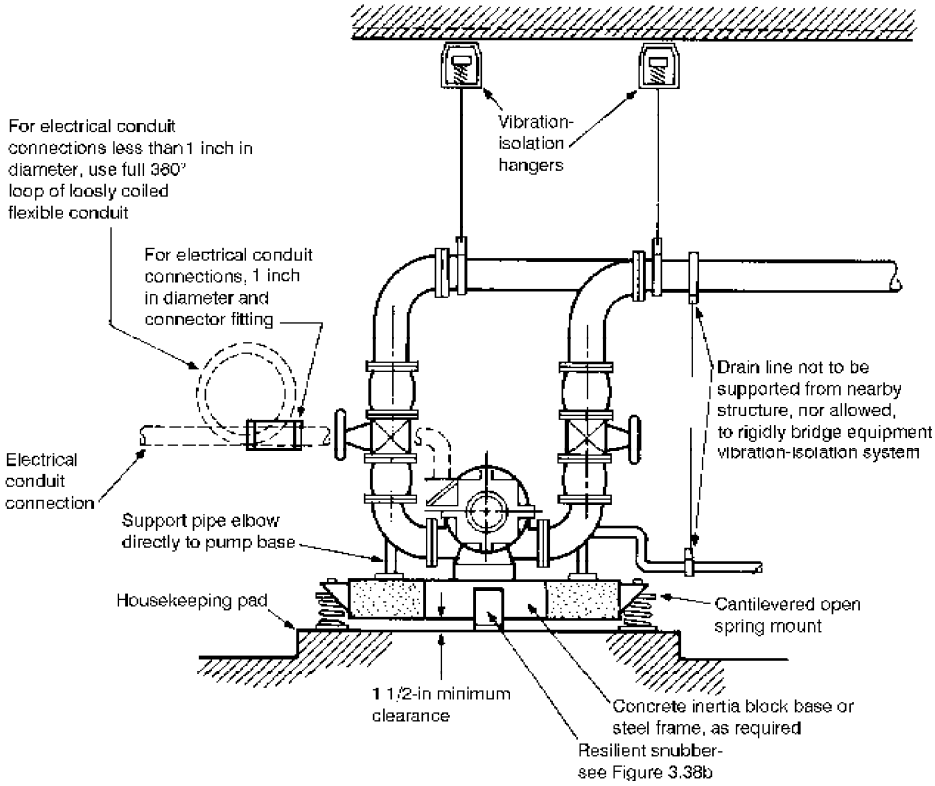

For electrical conduit connections less than 1 inch in diameter, use full 360° loop of loosly coiled flexible conduit
Vibration-isolation hangers
For electrical conduit connections, 1 inch in diameter and connector fitting
Drain line not to be supported from nearby structure, nor allowed, to rigidly bridge equipment vibration-isolation system
Electrical conduit connection
Support pipe elbow directly to pump base
Housekeeping pad
Cantilevered open spring mount
1 1/2-in minimum clearance
Concrete inertia block base or steel frame, as required
Resilient snubber-see Figure 3.38b

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FIGURE 11
PROPERLY INSTALLED VIBRATION-ISOLATION PIPE HANGER
(Ref. 7)

Vibration-Isolation Hanger

The spring-in-series hanger with a neoprene element shown in Figure 3.105 is the proper type and is correctly installed. The isolator is clear of all nearby building construction and other utility systems. The unit may be easily inspected and adjusted; when installed correctly, it will function adequately.

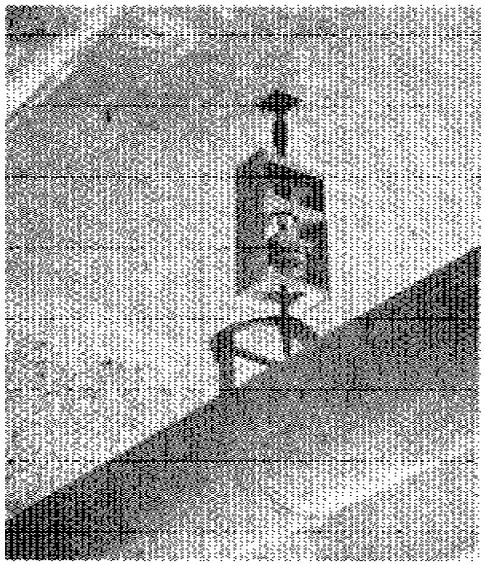

Fig. 3.105 Vibration-isolation hanger.

- **Rules of Thumb When Specifying Vibration Isolators -**

1. When specifying isolators for general rotating machinery, always ensure that the lowest significant forcing frequency is at least 3 times higher than the isolator natural frequency (in order to provide approximately 90% isolation). For more precision systems where more isolation is required, this ratio should be increased up to approximately 6 or 7.

2. Never specify an isolator having a natural frequency more than one-half the lowest significant frequency (don't forget the "belt RPM" in belt-driven systems; for example, in a 1780 RPM motor driving a 1000 RPM fan, the belt speed might be as low as 250 to 300 RPM which might prove to be a problem if five or six belts are required to drive the fan and problems develop with the belts). Therefore, it is always a good idea to take an analyzer into the field and capture a series of spectra, not only on the machine which is to be isolated, but also on and around other nearby machinery which might have even lower frequency vibration.

3. If isolating a direct coupled motor and pump, or a belt-driven fan, the analyst should consider installing a common base to support each of them (like that shown in Figure 8). Not doing so can introduce instability allowing each machine component to rock relative to one another.

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CHAPTER 11

INTRODUCTION TO DAMPING TREATMENTS

Dynamic responses of machines and structures are primarily governed by three properties - mass, stiffness and damping. The first two of these are most often readily understood by the vibration analyst. However, the third property, damping, is probably the most elusive and seemingly complex property of these three and is definitely the most difficult to predict. Unfortunately, the word "damping" has very loosely been used for years to apply to any number of vibration and noise reduction problems. That is, this term (damping) is frequently used with almost self-contradictory phrases such as "damp out the sound in a room", or "we use damping mounts under our machines". This indicates a state of popular confusion between the mechanisms of vibration damping, vibration isolation and sound absorption. Therefore, the first item we must do is to clearly define what is meant by "damping":

- **Damping -** Damping is the dissipation of vibratory energy with motion or with time, converting this energy from mechanical motions to other types of energy, chiefly heat. Therefore, the measurement of damping of a structure or material reveals its ability to rid itself or disperse vibration over time and distance. The lower the damping, the longer it will require a component to come back to rest once it has been impacted by an external force.

  Figure 1 helps illustrate the most important function of damping. Figure 1 shows three distinctly different regions (A, B and C) which shows the major limiting factor to vibration in each of these three regions. Region A shows a rotor operating below resonance where vibration due to unbalance will increase with the square of speed. Note that the dominant resistance to vibration in Region A will be the stiffness of the support system where stiffness, K, is expressed in units of pounds per inch (lb/in). Therefore, the amount of damping within the rotor has only negligible influence on resultant vibration when the rotor is operating within Region A.

  However, when the rotor accelerates and begins to approach resonance in Region B, the major limiting factor to vibration is the amount of damping within the system (where the "system" can include the rotor, supporting bearings and even the support structure). When the rotor goes into resonance, the only limiting factor to infinite vibration amplitude is the amount of damping within the system. In fact, lightly damped systems can easily experience from 20 to 100X the amount of vibration when at resonance, again limited only by the amount of damping. These reduced vibrations due to greater damping in turn will result in lowering of stress, fatigue and sound radiation.

  Figure 1 next shows that if the rotor is accelerated beyond the resonant amplification Region B, it will pass into Region C where both system damping and stiffness begin to have less and less influence on amplitude response. When fully into Region C, the rotor will respond almost as a pure mass, with almost no resistance supplied from spring stiffness or from damping.

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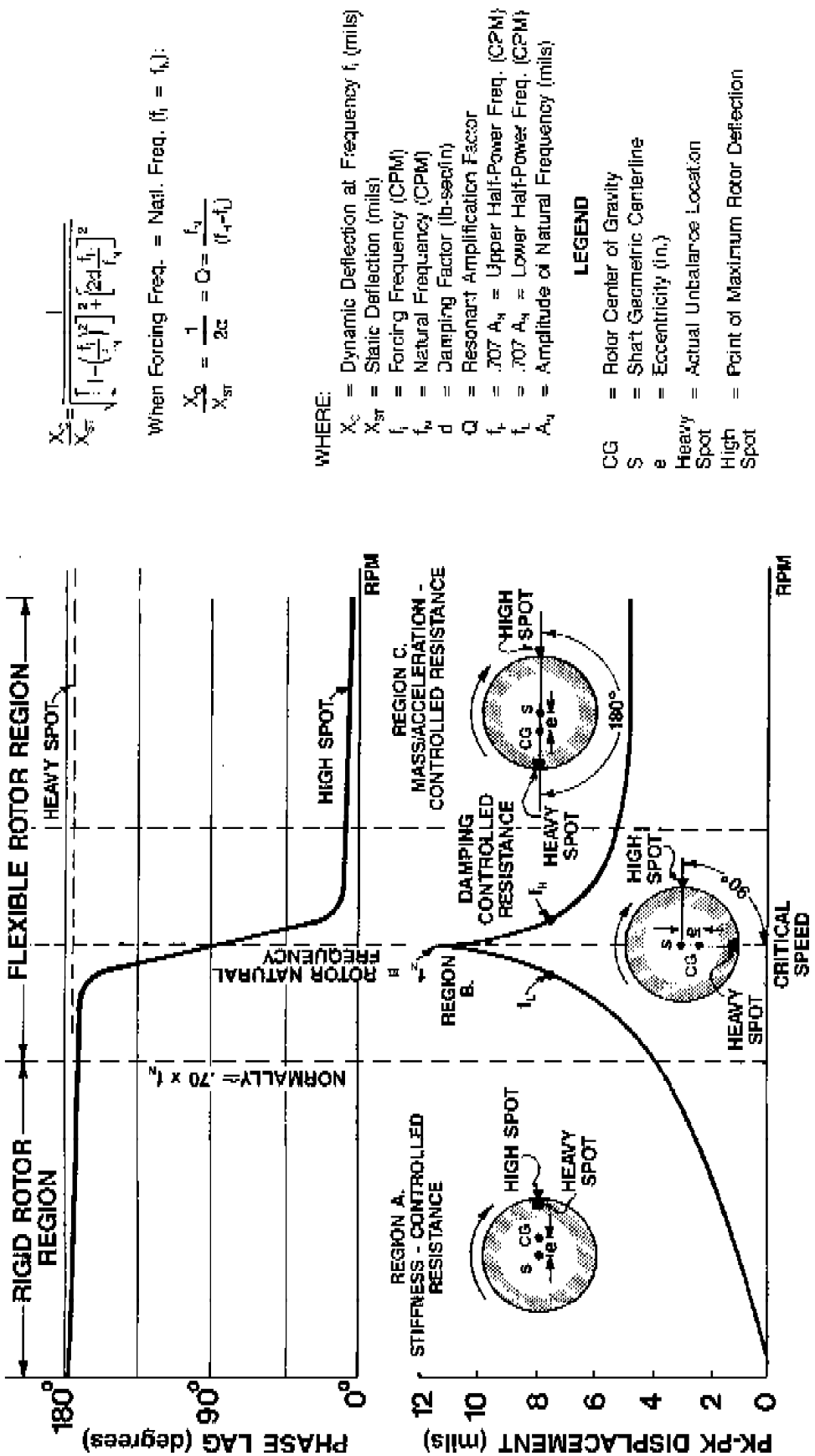

FIGURE 1
CHANGE OF VIBRATION DISPLACEMENT AND PHASE LAG WITH RPM ABOVE, BELOW, AND AT ROTOR RESONANCE (Ref. 1)

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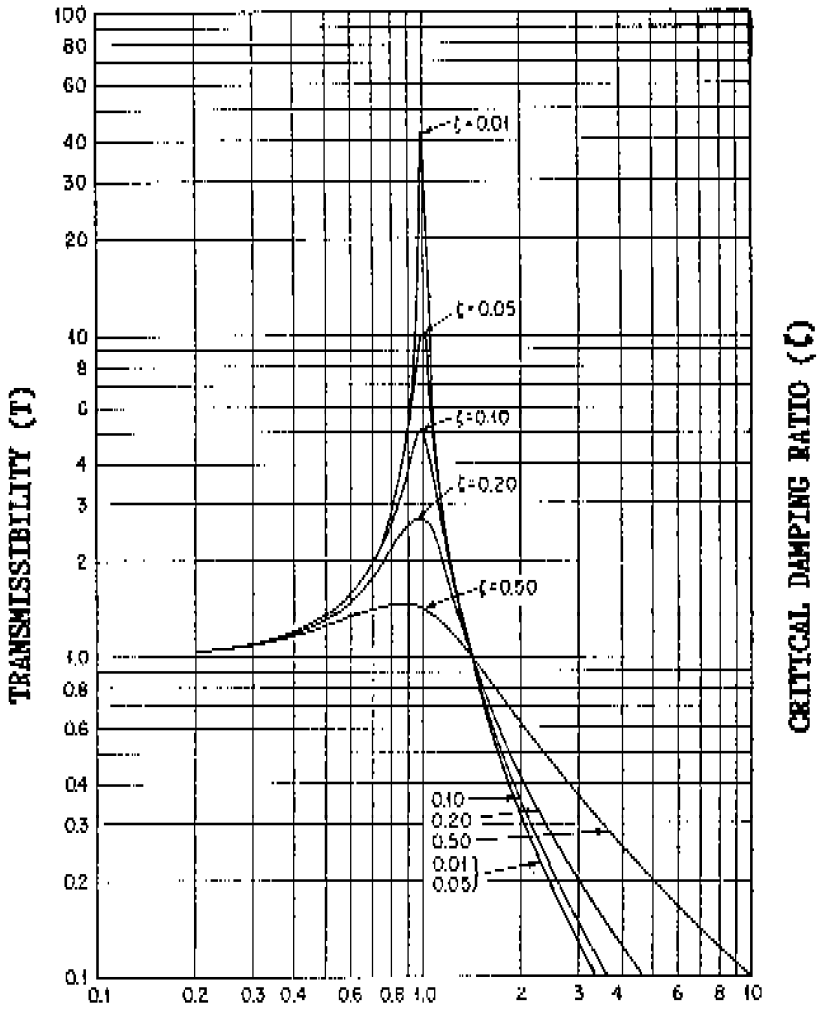
FIGURE 2
THEORETICAL TRANSMISSIBILITY CURVE FOR A RANGE OF DAMPING FACTORS (z) IN A SINGLE DEGREE-OF-FREEDOM SYSTEM (Ref. 2)



$$\frac{f_d}{f_n} = \frac{\text{Driving Frequency}}{\text{Natural Frequency}}$$

$$\text{TRANSMISSIBILITY} = T = \sqrt{\frac{1 + \left[(2\zeta)\left(\frac{f_d}{f_n}\right)\right]^2}{\left[1 - \left(\frac{f_d}{f_n}\right)^2\right]^2 + \left[(2\zeta)\left(\frac{f_d}{f_n}\right)\right]^2}}$$

Where:

T = Transmissibility
f_d = Driving or Forcing Frequency (Hz or CPM)
f_n = Natural Frequency (Hz or CPM)
$\zeta = c/c_c$ = Critical Damping Ratio = $\frac{1}{2Q}$
Q = Resonant Amplification Factor = $\frac{1}{2\zeta}$

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Figure 2 shows how a machine or structure is effected by the amount of damping when operating at resonance. The vertical scale labeled "Transmissibility" (T) describes how much vibration is amplified when operating below, at resonance and when passing through resonance. The horizontal scale is the ratio of the driving frequency (f_d) divided by the natural frequency (f_n). Therefore, when the driving frequency equals the natural frequency, this is the condition of resonance. The equation given on Figure 2 for transmissibility (T) includes the damping term known as the "critical damping ratio (ζ) which is also sometimes referred to as the "damping factor". Note on Figure 2 that the resonant amplification factor (Q) is also shown and equals 1 divided by 2X the damping factor (see Equation 1). This is proven by looking back at the transmissibility equation and setting the driving frequency (f_d) equal to the natural frequency (f_n). Therefore, when at resonance, the driving frequency coincides with the natural frequency. This driving frequency might be the operating speed, but can be any frequency which coincides with any natural frequency. That is, it might be the blade pass frequency (#blades X RPM) of a pump which lines up with a natural frequency; or a gear mesh frequency (#teeth X RPM), or simply 2X or 3X RPM. If any of these frequencies happen to line up with a system natural frequency, resonance will occur and vibration will be amplified. Typically, there is less and less energy with higher and higher multiples of operating speed so that the resonant vibration response is lower and lower as the higher harmonics of running speed pass through resonance. The important fact here is that numerous forcing frequencies can coincide with a natural frequency to cause resonance (not just operating speed frequency).

Equation 1 helps demonstrate the various forms used to express the amount of damping and shows how it can be calculated. Equation 1 also shows how damping relates to the resonant amplification factor (Q):

$$Q = \frac{1}{2\zeta} = \frac{1}{\eta} = \frac{\pi}{\delta} = \frac{f_n}{f_H - f_L} \qquad \text{EQUATION 1}$$

where:

- Q = Resonant Amplification Factor
- ζ = Critical Damping Ratio = c/c_c where c is the viscous damping coefficient (lb-sec/in) and c_c is the critical damping coefficient, or the smallest amount of damping required to return the system to its rest position without vacillation (lb-sec/in).
- η = 2ζ = Loss Factor for Damping Material which is the ratio of energy dissipated per radian to the maximum potential energy. It is a measure of the capacity of a vibration damping material to dissipate vibration.
- δ = Log Decrement Damping Factor which is a measure of the amount of damping based on the rate of decay of free oscillatory motion.
- f_n = Natural Frequency (Hz or CPM)
- f_H = Higher Half-Power Frequency just above f_n with an amplitude equal to .707X f_n amplitude (Hz or CPM)
- f_L = Lower Half-Power Frequency just below f_n with an amplitude equal to .707X f_n amplitude (Hz or CPM)

The importance of the resonant amplification factor (Q) is that Q shows how much higher the vibration response is since the machine is at resonance (correspondingly, it also shows how much lower the vibration could be if the machine were not operating at resonance at this particular frequency). To clarify this, Figure 3 is an actual spectrum captured when attempting to bring up a variable-speed machine to full operating speed (the machine was "supposed" to operate successfully up to 8000 RPM, but did not at that time). Looking at Figure 3, note there

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were several regions of higher vibration response including those at 2730 CPM (45.4 Hz), 4250 CPM (70.8 Hz) and 6525 CPM (108.75 Hz). What Figure 3 showed was that operation of the machine either close to 2730 or 6525 RPM would result in high vibration response. Suppose it was desired to operate the machine at 6525 RPM. Figure 3 shows that it would be subjected to a velocity of .415 in/sec which was clearly excessive for this particular machine. Therefore, a calculation of the resonant amplification factor (Q) as well as the damping ratio (ζ) was made on Figure 3. It used the items in Equation (1) to calculate Q where the natural frequency (f_n) was 6525 CPM; the lower half-power frequency (f_L) was 6350 CPM where the amplitude was 70.7% of that at f_n which is the definition of a half-power point; and the higher half-power frequency (f_H) was at 6725 CPM. Then, Figure 3 shows that the amplification factor (Q) was found to be 17.9. This means that resonant amplification was 17.9X higher than it would be if the machine were not resonant at this frequency (in other words, if this natural frequency could be changed by altering mass or stiffness moving it away from 6525 CPM, the resulting amplitude could be almost 18X lower, or .023 in/sec). Figure 3 then shows the calculation of the damping ratio (ζ) which was found to be .056.

Figure 3 helps make two other important points - (1) machines and structures commonly have several natural frequencies (in fact, they most always have a different set of natural frequencies in the horizontal, vertical and axial directions; as well as in each of three “rocking” directions); and (2) each natural frequency has its own specific damping factor (ζ). Looking back at Figure 3, note how much higher the response was at 2730 CPM than it was at 4250 (and likewise higher than at 6525 CPM). Also, note how much wider the resonant response curve was at 6525 CPM than it was at 2730 CPM. This means that the 2730 CPM resonance had a lower damping factor than that at 6525 CPM; and therefore, had higher resonant vibration response.

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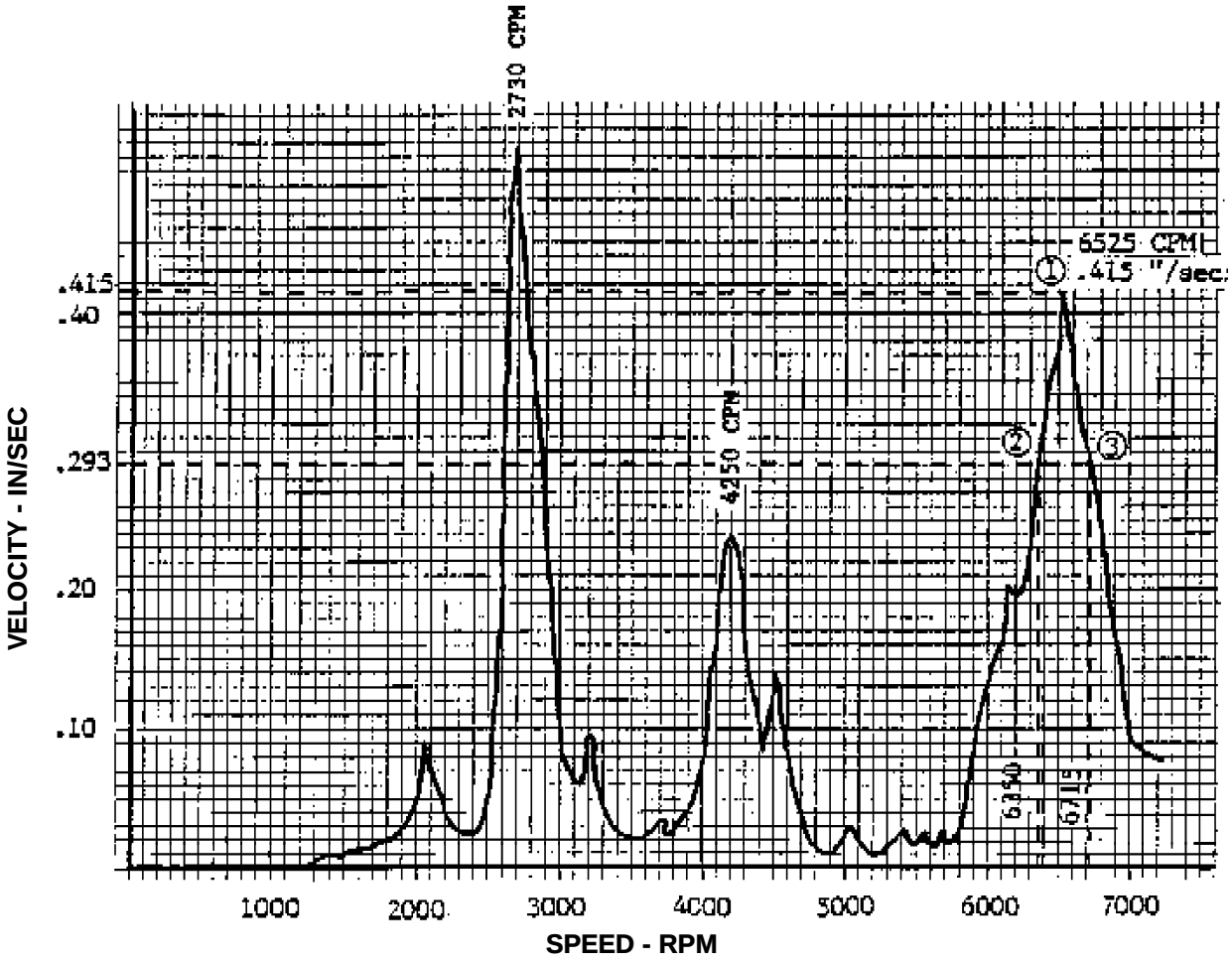
FIGURE 3
DAMPING RATIO FROM COASTDOWN VIBRATION



1 - At peak amplitude of .415 "/sec the frequency is 6525 CPM

2 - The half power points are .415 x .707 = .293 "/sec.

3 - The frequencies at the half power points are 6350 CPM and 6715 CPM

4 - The amplification

$$Q = \frac{6525}{6715 - 6350} = 17.9$$

5 - The damping ratio:

$$Q = \frac{1}{\sqrt{\left[1-\left(\frac{\omega}{\omega n}\right)^2\right]^2 + \left(\frac{c}{cc}\frac{\omega}{\omega n}\right)^2}} \qquad \text{at resonance} \qquad Q = \frac{1}{2\frac{c}{cc}\frac{\omega}{\omega n}}$$

$$\zeta = \frac{c}{cc} = \frac{1}{2Q} = \frac{1}{2(17.9)} = .056$$

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Of course, resonance is something which should be avoided by all machines or structures. However, some machines cannot avoid operation at resonance. For example, there are many machines such as rolling mills or winders which commonly are forced to pass through several resonances during operation. For example, note the synthetic fibers winder shown in Figure 4 which must maintain a constant surface speed (ft/min or m/min) throughout the winding process. Note that the winder is made up of a drive roll which comes down and contacts the chuck spindle which normally will carry anywhere from 2 to 4 bobbins on which synthetic yarn filament is wound. While the drive roll maintains a constant rotating speed throughout the winding operation, the chuck spindle carrying the bobbins constantly decreases in speed as more and more yarn is wound up on the bobbins. Therefore, most all of these types of machines are forced to run though one or more natural frequencies as not only the chuck spindle's speed (RPM) decreases, but also the spindle mass increases up to a full package which commonly might range from 20 to 30 pounds per bobbin. In these cases, the chuck spindle may initially pass through a vertical natural frequency when it is roughly half full; then it might pass through a horizontal natural frequency when it slows down when approaching full package size. Unfortunately, with increasing package size, not only might the chuck spindle be forced into resonance, but it very likely will have higher and higher package eccentricity as it winds more and more product (even though intense measures are made to prevent the filament from laying one strand on top of one another in a process called "ribbon breaking" in the synthetic fiber industry). Therefore, these machines will often suffer excessive vibration when they pass through these resonances, particularly if this occurs near full package size.

In the case of machines like that shown in Figure 4 which must pass through one or more resonances often suffering excessive vibration, corrective measures must be taken to help reduce the amount of resonant vibration response. As this section will show, this can be accomplished by increasing the damping of the system by use of a variety of damping treatments.

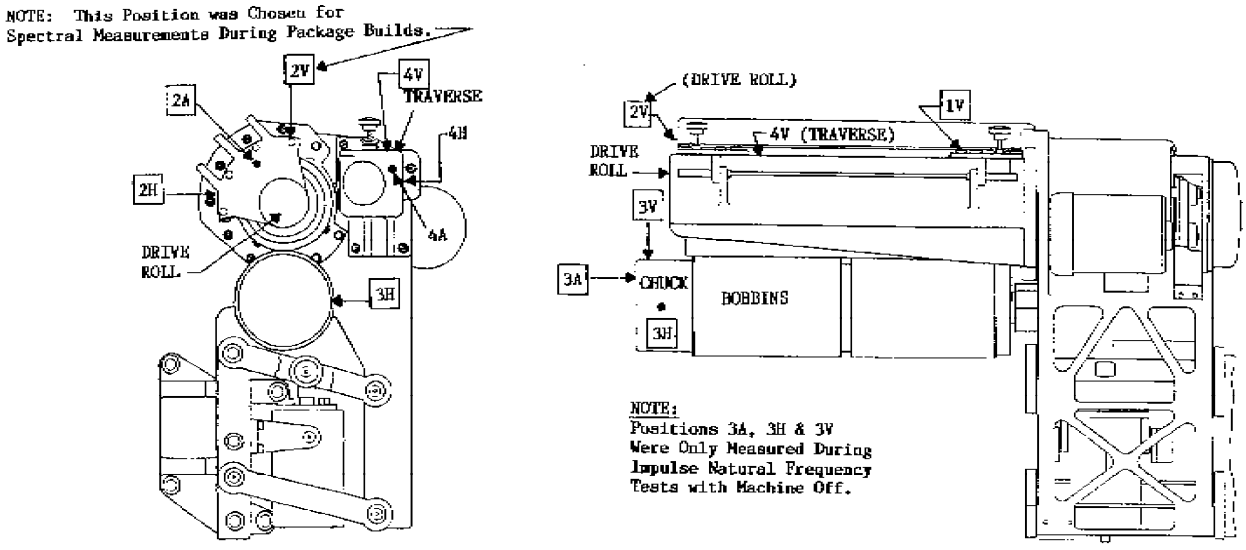

FIGURE 4
MEASUREMENT LOCATIONS FOR SYNTHETIC FIBERS WINDER

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- **Types of Damping Treatments -** There are two major types of surface damping treatments. These include the free-layer and the constrained-layer type. Figure 5 shows the free-layer treatment which consists of a single layer of viscoelastic material applied to the base structure, and depends on bending vibration of the base structure onto which it is applied to induce deformation of the viscoelastic material. That is, it depends on the metal on which it is placed to extend and to compress it with each vibration cycle. In so doing, energy is dissipated as a result of extension and compression of the material under flexural stress from the base structure. In the case of Figure 5, note that a constant thickness damping material (3/16 in) was applied to each of five different steel plates ranging in thickness from 1/32 inch up to 1/2 inch thick. Figure 5 shows how this effected the system loss factor (η) which is defined by Equation (1). Figure 5 shows that the damping treatment had greater and greater effect with decreasing steel plate thickness. Figure 5 also shows that there was an optimum temperature where the loss factor was highest, and that it changed from one steel plate thickness to another. This is common with damping materials. In fact, when choosing damping treatments, a whole series of treatment types can be used.

  Figure 6 shows performance data for a number of damping materials provided by various vendors. Note how they each have their own set of curves graphing the damping loss factor (η) versus frequency. Notice how much their damping capabilities can vary with temperature. For example looking at one of the damping treatments labelled EAR C-2003, this material is nominally effective at the frequency of 400 Hz (24,000 CPM) when the temperature of the surface onto which the material has been applied is either roughly 40°F to 60°F and at 140°F to 160°F (where the loss factor, η = .60). It would provide maximum damping at temperatures of 75° to 100°F (where the loss factor = 1.0). On the other hand, this material would provide little damping at temperatures either below 25°F or above 160°F when the loss factor fell below .20. For high temperature surfaces of about 250°F at this 400 Hz frequency, Figure 6 shows a better choice would likely be the Soundcoat DYAD 609.

  In general, to provide optimum damping, one must first determine the predominant frequencies present. Then, he must determine the appropriate temperatures to which the damping treatments will be subjected. Once he knows this, by using a chart like that shown in Figure 6, he should then select the optimum damping treatment under these operating conditions. Another important point is that when placing the damping treatments on the structure, they should be located where the vibration displacement is highest in order to be most effective.

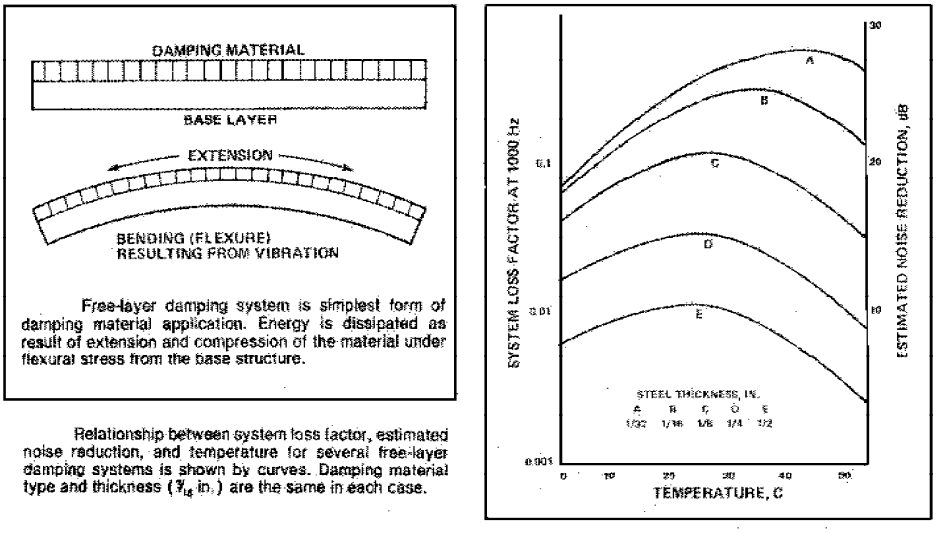
DAMPING MATERIAL
BASE LAYER
EXTENSION
BENDING (FLEXURE) RESULTING FROM VIBRATION
Free-layer damping system is simplest form of damping material application. Energy is dissipated as result of extension and compression of the material under flexural stress from the base structure.
Relationship between system loss factor, estimated noise reduction, and temperature for several free-layer damping systems is shown by curves. Damping material type and thickness (3/16 in.) are the same in each case.
SYSTEM LOSS FACTOR AT 1000 Hz
ESTIMATED NOISE REDUCTION, dB
STEEL THICKNESS, IN.
A B C D E
1/32 1/16 1/8 1/4 1/2
TEMPERATURE, C

FIGURE 5
FREE LAYER DAMPING (Ref. 3)

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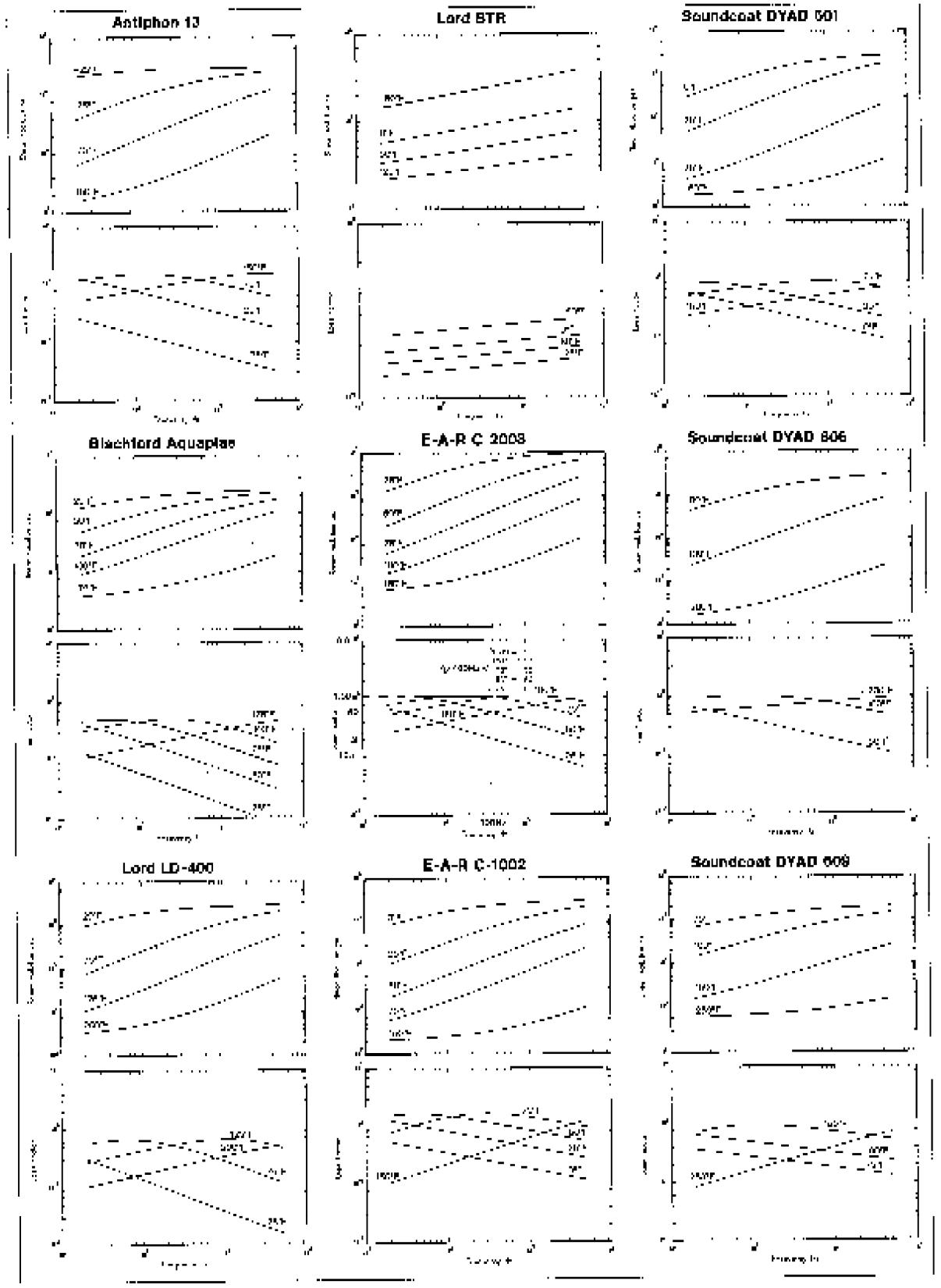

FIGURE 6
PHYSICAL PROPERTIES OF VARIOUS COMMERCIAL DAMPING MATERIALS
(From Anatrol Corp. materials properties data base.) (Ref. 4)

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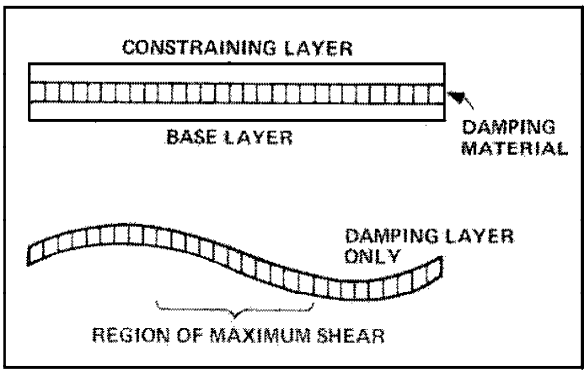
Figure 7 shows the other major type of damping called “constrained-layer damping”. In this case, the constrained-layer treatment consists of two layers - one being a layer of viscoelastic material, along with a layer of relatively stiff material (usually sheet metal). Each of these are bonded to the machine surface. Now, the bending vibration of the base structure induces shear deformation of the viscoelastic material due to the constraint provided on the outer boundary of the viscoelastic layer of the added layer of steel material. Therefore, one of the key parameters in damping efficiency will be determined by the adhesive bonding of the damping material to the base layer and to the constraining layer. This adhesive must have high shear stiffness. The shear strain in the adhesive will reduce the strain in the damping layer which will reduce its overall effectiveness. In general, constrained-layered damping systems will be much more efficient than free-layer treatments since vibration energy will be dissipated not only through deformation of the damping material, but also of the adhesive in constraining layers. In addition, constrained-layer damping often adds strength to the structure which can help lower vibration even in the regions below the natural frequency since this region will predominantly be stiffness controlled. Here again, Figure 7 shows that the constrained-layer treatments will likewise be temperature dependant having their certain optimum temperature and frequency regions in which they will be most effective.



Constrained-layer damping system is usually recommended for stiff, structural applications. Energy is dissipated as a result of shear deformation of the damping layer.

Relationship between system loss factor, estimated noise reduction, and temperature for several constrained-layer damping systems is shown by curves. Damping material type and thickness ($^3/_{16}$ in.) are the same in each case.

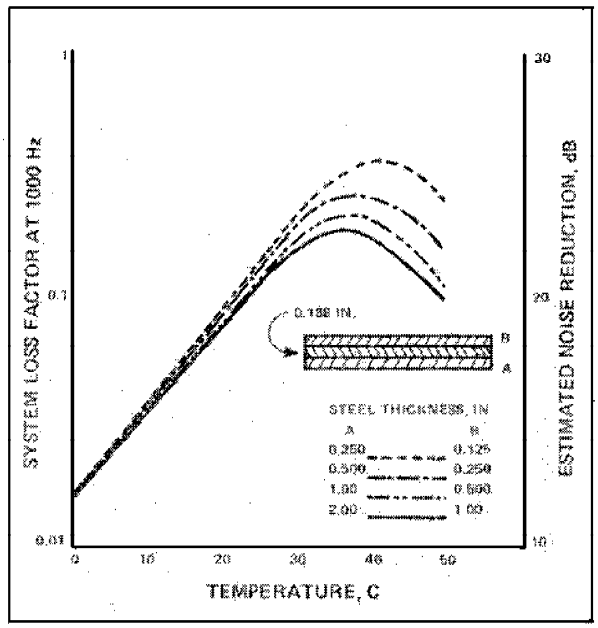


FIGURE 7
CONSTRAINED LAYER DAMPING (Ref. 3) EFFECTS OF TEMPERATURE, FREQUENCY AND THICKNESS ON THE RESULTANT SYSTEM DAMPING

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Glossary

REFERENCE: **Hewlett Packard DYNAMIC SIGNAL ANALYZER APPLICATIONS; "Effective Machinery Measurements Using Dynamic Signal Analyzers, Applications Notes 243-1; Oct. 1991.**

A

Acceleration. The time rate of change of velocity. Typical units are ft/s/s, meters/s/s, and G's (1G = 32.17 ft/s/s = 9.81 m/s/s). Acceleration measurements are usually made with accelerometers.

Accelerometer. Transducer whose output is directly proportional to acceleration. Most commonly use piezoelectric crystals to produce output.

Aliasing. A phenomenon which can occur whenever a signal is not sampled at greater than twice the maximum frequency component. Causes high frequency signals to appear at low frequencies. Aliasing is avoided by filtering out signals greater than 1/2 the sample rate.

Alignment. A condition whereby the axes of machine components are either coincident, parallel or perpendicular, according to design requirements.

Amplification Factor (Synchronous). A measure of the susceptibility of a rotor to vibration amplitude when rotational speed is equal to the rotor natural frequency (implies a flexible rotor). For imbalance type excitation, synchronous amplification factor is calculated by dividing the amplitude value at the resonant peak by the amplitude value at a speed well above resonance (as determined from a plot of synchronous response vs. rpm).

Amplitude. The magnitude of dynamic motion or vibration. Amplitude is expressed in terms of peak-to-peak, zero-to-peak, or rms. For pure sine waves only, these are related as follows: rms = 0.707 times zero-to-peak; peak-to-peak = 2 times zero-to-peak. DSAs generally read rms for spectral components, and peak for time domain components.

Anti-Aliasing Filter. A low-pass filter designed to filter out frequencies higher than 1/2 the sample rate in order to prevent aliasing.

Anti-Friction Bearing. See Rolling Element Bearing.

Asymetrical Support. Rotor support system that does not provide uniform restraint in all radial directions. This is typical for most heavy industrial machinery where stiffness in one plane may be substantially different than stiffness in the perpendicular plane. Occurs in bearings by design, or from preloads such as gravity or misalignment.

Asynchronous. Vibration components that are not related to rotating speed (also referred to as nonsynchronous).

Attitude Angle (Steady-State). The angle between the direction of steady-state preload through the bearing centerline, and a line drawn between the shaft centerline and the bearing centerline. (Applies to fluid film bearings.)

Auto Spectrum (Power Spectrum). DSA spectrum display whose magnitude represents the power at each frequency, and which has no phase. Rms averaging produces an auto spectrum.

Averaging. In a DSA, digitally averaging several measurements to improve accuracy or to reduce the level of asynchronous components. Refer to definitions of rms, time, and peak-hold averaging.

Axial. In the same direction as the shaft centerline.

Axial Position. The average position, or change in position, of a rotor in the axial direction with respect to some fixed reference position. Ideally the reference is a known position within the thrust bearing axial clearance or float zone, and the measurement is made with a displacement transducer observing the thrust collar.

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B

Balancing Resonance Speed(s). A rotative speed that corresponds to a natural resonance frequency.

Balanced Condition. For rotating machinery, a condition where the shaft geometric centerline coincides with the mass centerline.

Balancing. A procedure for adjusting the radial mass distribution of a rotor so that the mass centerline approaches the rotor geometric centerline.

Bandpass Filter. A filter with a single transmission band extending from lower to upper cutoff frequencies. The width of the band is determined by the separation of frequencies at which amplitude is attenuated by 3 dB (0.707).

Bandwidth. The spacing between frequencies at which a bandpass filter attenuates the signal by 3 dB. In a DSA, measurement bandwidth is equal to [(frequency span)/(number of filters) x (window factor)]. Window factors are: 1 for uniform, 1.5 for Hanning, and 3.63 for Flat Top.

Baseline Spectrum. A vibration spectrum taken when a machine is in good operating condition; used as a reference for monitoring and analysis.

Blade Passing Frequency. A potential vibration frequency on any bladed machine (turbine, axial compressor, fan, etc.). It is represented by the number of blades times shaft-rotating frequency.

Block Size. The number of samples used in a DSA to compute the Fast Fourier Transform. Also the number of samples in a DSA time display. Most DSAs use a block size of 1024. Smaller block size reduces resolution.

Bode'. Rectangular coordinate plot of 1x component amplitude and phase (relative to a keyphasor) vs. running speed.

BPFO, BPFI. Common abbreviations for ball pass frequency of defects on outer and inner bearing races, respectively.

Bow. A shaft condition such that the geometric centerline of the shaft is not straight.

Brinelling (False). Impressions made by bearing rolling elements on the bearing race; typically caused by external vibration when the shaft is stationary.

C

Calibration. A test during which known values of the measured variable are applied to the transducer or readout instrument, and output readings varied or adjusted.

Campbell Diagram. A mathematically constructed diagram used to check for coincidence of vibration sources (i.e. 1x imbalance, 2x misalignment) with rotor natural resonances. The form of the diagram is a rectangular plot of resonant frequency (y-axis) vs excitation frequency (x-axis). Also known as an interference diagram.

Cascade Plot. See Spectral Map.

Cavitation. A condition which can occur in liquid handling machinery (e.g. centrifugal pumps) where a system pressure decrease in the suction line and pump inlet lowers fluid pressure and vaporization occurs. The result is mixed flow which may produce vibration.

Center Frequency. For a bandpass filter, the center of the transmission band.

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Charge Amplifier. Amplifier used to convert accelerometer output impedance from high to low, making calibration much less dependent on cable capacitance.

Coherence. The ratio of coherent output power between channels in a dual-channel DSA. An effective means of determining the similarity of vibration at two locations, giving insight into the possibility of cause and effect relationships.

Constant Bandwidth Filter. A bandpass filter whose bandwidth is independent of center frequency. The filters simulated digitally in a DSA are constant band width.

Constant Percentage Bandwidth. A bandpass filter whose bandwidth is a constant percentage of center frequency. 1/3 octave filters, including those synthesized in DSAs, are constant percentage bandwidth.

Critical Machinery. Machines which are critical to a major part of the plant process. These machines are usually unspared.

Critical Speeds. In general, any rotating speed which is associated with high vibration amplitude. Often, the rotor speeds which correspond to natural frequencies of the system.

Critical Speed Map. A rectangular plot of system natural frequency (y-axis) versus bearing or support stiffness (x-axis).

Cross Axis Sensitivity. A measure of off-axis response of velocity and acceleration transducers.

Cycle. One complete sequence of values of a periodic quantity.

D

Damping. The quality of a mechanical system that restrains the amplitude of motion with each successive cycle. Damping of shaft motion is provided by oil in bearings, seals, etc. The damping process converts mechanical energy to other forms, usually heat.

Damping, Critical. The smallest amount of damping required to return the system to its equilibrium position without oscillation.

Decibels (dB). A logarithmic representation of amplitude ratio, defined as 20 times the base ten logarithm of the ratio of the measured amplitude to a reference. dBV readings, for example, are referenced to 1 volt rms. dB amplitude scales are required to display the full dynamic range of a DSA.

Degrees Of Freedom. A phrase used in mechanical vibration to describe the complexity of the system. The number of degrees of freedom is the number of independent variables describing the state of a vibrating system.

Digital Filter. A filter which acts on data after it has been sampled and digitized. Often used in DSAs to provide anti-aliasing protection after internal re-sampling.

Differentiation. Representation in terms of time rate of change. For example, differentiating velocity yields acceleration. In a DSA, differentiation is performed by multiplication by jw, where w is frequency multiplied by 2π (Differentiation can also be used to convert displacement to velocity.)

Discrete Fourier Transform. A procedure for calculating discrete frequency components (filters or lines) from sampled time data. Since the frequency domain result is complex (i.e., real and imaginary components), the number of points is equal to half the number of samples.

Displacement. The change in distance or position of an object relative to a reference.

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Displacement Transducer. A transducer whose output is proportional to the distance between it and the measured object (usually the shaft).

DSA. See Dynamic Signal Analyzer.

Dual Probe. A transducer set consisting of displacement and velocity transducers. Combines measurement of shaft motion relative to the displacement transducer with velocity of the displacement transducer to produce absolute motion of the shaft.

Dual Voting. Concept where two independent inputs are required before action (usually machine shutdown) is taken. Most often used with axial position measurements, where failure of a single transducer might lead to an unnecessary shutdown.

Dynamic Motion. Vibratory motion of a rotor system caused by mechanisms that are active only when the rotor is turning at speeds above slow roll speed.

Dynamic Signal Analyzer (DSA). Vibration analyzer that uses digital signal processing and the Fast Fourier Transform to display vibration frequency components. DSAs also display the time domain and phase spectrum, and can usually be interfaced to a computer.

E

Eccentricity, Mechanical. The variation of the outer diameter of a shaft surface when referenced to the true geometric centerline of the shaft. Out-of-roundness.

Eccentricity Ratio. The vector difference between the bearing centerline and the average steady-state journal centerline.

Eddy Current. Electrical current which is generated (and dissipated) in a conductive material in the presence of an electromagnetic field.

Electrical Runout. An error signal that occurs in eddy current displacement measurements when shaft surface conductivity varies.

Engineering Units. In a DSA, refers to units that are calibrated by the user (e.g., in/s, g's).

External Sampling. In a DSA, refers to control of data sampling by a multiplied tachometer signal. Provides a stationary display of vibration with changing speed.

F

Fast Fourier Transform (FFT). A computer (or microprocessor) procedure for calculating discrete frequency components from sampled time data. A special case of the discrete Fourier transform where the number of samples is constrained to a power of 2.

Filter. Electronic circuitry designed to pass or reject a specific frequency band.

Finite Element Modeling. A computer aided design technique for predicting the dynamic behavior of a mechanical system prior to construction. Modeling can be used, for example, to predict the natural frequencies of a flexible rotor.

Flat Top Filter. DSA window function which provides the best amplitude accuracy for measuring discrete frequency components.

Fluid-Film Bearing. A bearing which supports the shaft on a thin film of oil. The fluid-film layer may be generated by journal rotation (hydrodynamic bearing), or by externally applied pressure (hydrostatic bearing).

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Forced Vibration. The oscillation of a system under the action of a forcing function. Typically forced vibration occurs at the frequency of the exciting force.

Free Vibration. Vibration of a mechanical system following an initial force-typically at one or more natural frequencies.

Frequency. The repetition rate of a periodic event, usually expressed in cycles per second (Hz), revolutions per minute (rpm), or multiples of a rotational speed (orders). Orders are commonly referred to as 1x for rotational speed, 2x for twice rotational speed, etc.

Frequency Response. The amplitude and phase response characteristics of a system.

G

G. The value of acceleration produced by the force of gravity.

Gear Mesh Frequency. A potential vibration frequency on any machine that contains gears; equal to the number of teeth multiplied by the rotational frequency of the gear.

H

Hanning Window. DSA window function that provides better frequency resolution than the flat top window, but with reduced amplitude accuracy.

Harmonic. Frequency component at a frequency that is an integer multiple of the fundamental frequency.

Heavy Spot. The angular location of the imbalance vector at a specific lateral location on a shaft. The heavy spot typically does not change with rotational speed.

Hertz (Hz). The unit of frequency represented by cycles per second.

High Spot. The angular location on the shaft directly under the vibration transducer at the point of closest proximity. The high spot can move with changes in shaft dynamics (e.g., from changes in speed).

High-Pass Filter. A filter with a transmission band starting at a lower cutoff frequency and extending to (theoretically) infinite frequency.

Hysteresis. Non-uniqueness in the relationship between two variables as a parameter increases or decreases. Also called deadband, or that portion of a system's response where a change in input does not produce a change in output.

I

Imbalance. Unequal radial weight distribution on a rotor system; a shaft condition such that the mass and shaft geometric centerlines do not coincide.

Impact Test. Response test where the broad frequency range produced by an impact is used as the stimulus. Sometimes referred to as a bump test.

Impedance, Mechanical. The mechanical properties of a machine system (mass, stiffness, damping) that determine the response to periodic forcing functions.

Influence Coefficients. Mathematical coefficients that describe the influence of system loading on system deflection.

Integration. A process producing a result that, when differentiated, yields the original quantity. Integration of acceleration, for example, yields velocity. Integration is performed in a DSA by dividing by jw, where w is frequency multiplied by 2π. (Integration is also used to convert velocity to displacement).

J

Journal. Specific portions of the shaft surface from which rotor applied loads are transmitted to bearing supports.

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K

Keyphasor. A signal used in rotating machinery measurements, generated by a transducer observing a once-per-revolution event. The keyphasor signal is used in phase measurements for analysis and balancing. (Keyphasor is a Bently Nevada trade name.)

L

Lateral Location. The definition of various points along the shaft axis of rotation.

Lateral Vibration. See Radial Vibration.

Leakage. In DSAs, a result of finite time record length that results in smearing of frequency components. Its effects are greatly reduced by the use of weighted window functions such as Flat Top and Hanning.

Linearity. The response characteristics of a linear system remain constant with input level. That is, if the response to input a is A, and the response to input b is B, then the response of a linear system to input (a + b) will be (A + B). An example of a non-linear system is one whose response is limited by mechanical stop, such as occurs when a bearing mount is loose.

Lines. Common term used to describe the filters of a DSA (e.g., 400 line analyzer).

Linear Averaging. See Time Averaging.

Low-Pass Filter. A filter whose transmission band extends from dc to an upper cutoff frequency.

M

Mechanical Runout. An error in measuring the position of the shaft centerline with a displacement probe that is caused by out-of-roundness and surface imperfections.

Micrometer (MICRON). One millionth (.000001) of a meter. (1 micron = 1 x E-6 meters = 0.04 mils.)

MIL. One thousandth (0.001) of an inch. (1 mil = 25.4 microns.)

Modal Analysis. The process of breaking complex vibration into its component modes of vibration, very much like frequency domain analysis breaks vibration down to component frequencies.

Mode Shape. The resultant deflected shape of a rotor at a specific rotational speed to an applied forcing function. A three-dimensional presentation of rotor lateral deflection along the shaft axis.

Modulation, Amplitude (AM). The process where the amplitude of a signal is varied as a function of the instantaneous value of another signal. The first signal is called the carrier, and the second signal is called the modulating signal. Amplitude modulation produces a component at the carrier frequency, with adjacent components (sidebands) at the frequency of the modulating signal.

Modulation, Frequency (FM). The process where the frequency of the carrier is determined by the amplitude of the modulating signal. Frequency modulation produces a component at the carrier frequency, with adjacent components (sidebands) at the frequency of the modulating signal.

N

Natural Frequency. The frequency of free vibration of a system. The frequency at which an undamped system with a single degree of freedom will oscillate upon momentary displacement from its rest position.

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Nodal Point. A point of minimum shaft deflection in a specific mode shape. May readily change location along the shaft axis due to changes in residual imbalance or other forcing function, or change in restraint such as increased bearing clearance.

Noise. Any component of a transducer output signal that does not represent the variable intended to be measured.

Nyquist Criterion. Requirement that a sampled system sample at a frequency greater than twice the highest frequency to be measured.

Nyquist Plot. A plot of real versus imaginary spectral components that is often used in servo analysis. Should not be confused with a polar plot of amplitude and phase of 1x vibration.

O

Octave. The interval between two frequencies with a ratio of 2 to 1.

Oil Whirl/Whip. An unstable free vibration whereby a fluid-film bearing has insufficient unit loading. Under this condition, the shaft centerline dynamic motion is usually circular in the direction of rotation. Oil whirl occurs at the oil flow velocity within the bearing, usually 40 to 49% of shaft speed. Oil whip occurs when the whirl frequency coincide with (and becomes locked to) a shaft resonant frequency. (Oil whirl and whip can occur in any case where fluid is between two cylindrical surfaces.)

Orbit. The path of the shaft centerline motion during rotation. The orbit is observed with an oscilloscope connected to x and y-axis displacement transducers. Some dual-channel DSAs also have the ability to display orbits.

Oscillator-Demodulator. A signal conditioning device that sends a radio frequency signal to an eddy-current displacement probe, demodulates the probe output, and provides output signals proportional to both the average and dynamic gap distances. (Also referred to as Proximitor, a Bently Nevada trade name.)

P

Peak Hold. In a DSA, a type of averaging that holds the peak signal level for each frequency component.

Period. The time required for a complete oscillation or for a single cycle of events. The reciprocal of frequency.

Phase. A measurement of the timing relationship between two signals, or between a specific vibration event and a keyphasor pulse.

Piezoelectric. Any material which provides a conversion between mechanical and electrical energy. For a piezoelectric crystal, if mechanical stresses are applied on two opposite faces, electrical charges appear on some other pair of faces.

Polar Plot. Polar coordinate representation of the locus of the 1x vector at a specific lateral shaft location with the shaft rotational speed as a parameter.

Power Spectrum. See Auto Spectrum.

Preload, Bearing. The dimensionless quantity that is typically expressed as a number from zero to one where a preload of zero indicates no bearing load upon the shaft, and one indicates the maximum preload (i.e., line contact between shaft and bearing).

Preload, External. Any of several mechanisms that can externally load a bearing. This includes ‘ soft” preloads such as process fluids or gravitational forces as well as “hard” preloads from gear contact forces, misalignment, rubs, etc.

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R

Radial. Direction perpendicular to the shaft centerline.

Radial Position. The average location, relative to the radial bearing centerline, of the shaft dynamic motion.

Radial Vibration. Shaft dynamic motion or casing vibration which is in a direction perpendicular to the shaft centerline.

Real-Time Analyzer. See Dynamic Signal Analyzer.

Real-Time Rate. For a DSA, the broadest frequency span at which data is sampled continuously. Real-time rate is mostly dependent on FFT processing speed.

Rectangular Window. See Uniform Window.

Relative Motion. Vibration measured relative to a chosen reference. Displacement transducers generally measure shaft motion relative to the transducer mounting.

Repeatability. The ability of a transducer or readout instrument to reproduce readings when the same input is applied repeatedly.

Resolution. The smallest change in stimulus that will produce a detectable change in the instrument output.

Resonance. The condition of vibration amplitude and phase change response caused by a corresponding system sensitivity to a particular forcing frequency. A resonance is typically identified by a substantial amplitude increase, and related phase shift.

Rolling Element Bearing. Bearing whose low friction qualities derive from rolling elements (balls or rollers), with little lubrication.

Root Mean Square (rms). Square root of the arithmetical average of a set of squared instantaneous values. DSAs perform rms averaging digitally on successive vibration spectra.

Rotor, Flexible. A rotor which operates close enough to, or beyond its first bending critical speed for dynamic effects to influence rotor deformations. Rotors which cannot be classified as rigid rotors are considered to be flexible rotors.

Rotor, Rigid. A rotor which operates substantially below its first bending critical speed. A rigid rotor can be brought into, and will remain in, a state of satisfactory balance at all operating speeds when balanced on any two arbitrarily selected correction planes.

RPM Spectral Map. A spectral map of vibration spectra versus rpm.

Runout Compensation. Electronic correction of a transducer output signal for the error resulting from slow roll runout.

S

Seismic. Refers to an inertially referenced measurement or a measurement relative to free space.

Seismic Transducer. A transducer that is mounted on the case or housing of a machine and measures casing vibration relative to free space. Accelerometers and velocity transducers are seismic.

Signal Conditioner. A device placed between a signal source and a readout instrument to change the signal. Examples: attenuators, preamplifiers, charge amplifiers.

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Signature. Term usually applied to the vibration frequency spectrum which is distinctive and special to a machine or component, system or subsystem at a specific point in time, under specific machine operating conditions, etc. Used for historical comparison of mechanical condition over the operating life of the machine.

Slow Roll Speed. Low rotative speed at which dynamic motion effects from forces such as imbalance are negligible.

Spectral Map. A three-dimensional plot of the vibration amplitude spectrum versus another variable, usually time or rpm.

Spectrum Analyzer. An instrument which displays the frequency spectrum of an input signal.

Stiffness. The spring-like quality of mechanical and hydraulic elements to elasticity deform under load.

Strain. The physical deformation, deflection, or change in length resulting from stress (force per unit area).

Subharmonic. Sinusoidal quantity of a frequency that is an integral submultiple of a fundamental frequency.

Subsynchronous. Component(s) of a vibration signal which has a frequency less than shaft rotative frequency.

Synchronous Sampling. In a DSA, it refers to the control of the effective sampling rate of data; which includes the processes of external sampling and computed resampling used in order tracking.

T

Time Averaging. In a DSA, averaging of time records that results in reduction of asynchronous components.

Time Record. In a DSA, the sampled time data converted to the frequency domain by the FFT. Most DSAs use a time record of 1024 samples.

Torsional Vibration. Amplitude modulation of torque measured in degrees peak-to-peak referenced to the axis of shaft rotation.

Tracking Filter. A low-pass or bandpass filter which automatically tracks the input signal. A tracking filter is usually required for aliasing protection when data sampling is controlled externally.

Transducer. A device for translating the magnitude of one quantity into another quantity.

Transient Vibration. Temporarily sustained vibration of a mechanical system. It may consist of forced or free vibration or both. Typically this is associated with changes in machine operating condition such as speed, load, etc.

Transverse Sensitivity. See Cross-Axis Sensitivity.

Trigger. Any event which can be used as a timing reference. In a DSA, a trigger can be used to initiate a measurement.

U

Unbalance. See Imbalance.

Uniform Window. In a DSA, a window function with uniform weighting across the time record. This window does not protect against leakage, and should be used only with transient signals contained completely within the time record.

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Vector. A quantity which has both magnitude and direction (phase).

Waterfall Plot. See Spectral Map.

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ACTUAL CASE HISTORIES OF VIBRATION DIAGNOSTICS ON VARIOUS MACHINE TYPES

SECTION		SUBJECT	PAGE
I.	**UNBALANCE**		
	A.	Detection & Correction of Faults on an Air Handler Having Problems Due to Unbalance, Sheave Misalignment, Defective Belts, Resonance and a Cracked Frame Supporting the Outboard Fan Pillowblock Bearing	1
	B.	Detection & Correction of Unbalance on an Air Conditioner Supply Fan	12
II.	**ECCENTRIC ROTOR**		
	A.	Detection of Eccentric Orbit on a Turbine/Generator Due to Excessive Sleeve Bearing Wear	16
III.	**MISALIGNMENT**		
	A.	Effect of Coupling Misalignment on Vibration of a Water Scrubber Pump	23
IV.	**RESONANCE PROBLEMS**		
	A.	Resolution of a Severe Resonance Problem at 2X RPM Which Had Caused Continuing Impeller Shaft Fractures on a Certain Model of Vertical Pump	31
	B.	Correction of Excessive Vibration Transmission into Two XLE Reciprocating Air Compressors Due to Attached Piping Resonance	37
	C.	Detection of Resonant Whirl on Paper Machine Rolls	45
	D.	Detection and Correction of a Serious Rotor Resonance on a Centrifugal Compressor Which Had Successfully Operated for 15 Years Before a Recent Overhaul	52
	E.	How Results of Coastdown Natural Frequency Tests Were Able to Greatly Enhance the Quality and Open Up a New Market for a Machine with Serious Resonance Problems	68
V.	**MECHANICAL LOOSENESS/WEAKNESS**		
	A.	**Looseness Type A** - "Correction of an Incinerator Fan With a Loose Base and Unbalance"	79
	B.	**Looseness Type B** - "Unbalance & Looseness Problems on an In-Line Induced Draft Fan"	86
	C.	**Looseness Type C** - "Detection of Fan Wheel Rotor Looseness on a Nuva Feeder Vent Fan"	89
VI.	**ROTOR RUB**		
	A.	Diagnosis and Correction of a Serious Rub Problem on a Chemical Product Reactor Operating at 5 RPM to 60 RPM	93
	B.	Detection of Rotor Rub on a Lewis Screw Compressor	104
VII.	**SLEEVE BEARING PROBLEMS**		
	A.	Detection of Second Stage Plain Bearing Problems on a Centrifugal Air Compressor	110
	B.	Detection of Plain Bearing Problems in the Motor Serving a Centrifugal Air Compressor	115
VIII.	**ROLLING ELEMENT BEARING PROBLEMS**		
	A.	Detection of Serious Wear of a Centrifugal Compressor Bullgear SKF 6213 Ball Bearing	123
	B.	Detection of a Serious Rolling Element Bearing Fault on a Sullair Screw Compressor	133
	C.	Surprising Detection of a Rolling Element Bearing Problem on a Draw Frame Input Pinion Bearing After Replacement of a Falk Gearbox Which Had Been Thought to be the Source of the Problem (How It Pays to Know the Identity of Components within Your Critical Machinery)	143
	D.	Severe Vibration Trend of a Timken Rolling Bearing Installed in a Circulating Fan	158

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SECTION	SUBJECT	PAGE
VIII.	**ROLLING ELEMENT BEARING PROBLEMS (cont.)**	
E.	Serious Problem of a Bearing Turning on a Shaft of One Motor and Detection of Severe Bearing Problems on Another Motor as Evidenced by Numerous Outer Race Sum and Difference Frequencies	165
F.	Detection of a Ball Bearing Problem on a Gland Steam Exhaust Fan Whose Bearing Manufacturer and Model Number were Unknown	180
G.	"Electrically Induced High Frequency Problems on Rolling Element Bearings and What to Do About It" (As Per Update International and Robert Hoskins of R. R. Donnelley)	186
IX.	**FLOW INDUCED VIBRATION PROBLEMS**	
A.	Significant Hydraulic Force Vibration in a Boiler Feed Pump	188
B.	Cavitation Diagnosis within a Chilled Water Pump	196
C.	Distinguishing Between Pump Cavitation & Defective Bearings	202
X.	**GEAR PROBLEMS**	
A.	Detection of a Gear Wear Problem on a Paper Machine	210
B.	Cause of High Noise and Vibration Problems on a Draw Frame Bevel Gear Set	216
C.	Detection of Cracked Gear Teeth on a 56 RPM Calender Roll Using Time Waveform & Spectral Analysis	223
D.	Hunting Tooth Vibration Problem on a Gerotor Oil Pump Located on a Centrifugal Air Compressor	231
XI.	**ELECTRICAL PROBLEMS**	
A.	Detection of an Electrical Problem on a Primary Air Fan Motor	235
B.	Trending of a Motor Electrical Problem Which Eventually Concluded with Catastrophic Failure	241
C.	Rejection of a Centrifugal Chiller Motor During Acceptance Testing due to Noticeable Electrical Problems	243
D.	Detection of Loose Rotor Bars in a Dryer Exhaust Fan Motor	250
E.	Detection of Loose Rotor Bars in a Plough Shear Variable Speed AC Induction Motor	255
F.	Detection of Phasing Problem in a Variable Speed Chiller Motor	259
G.	Detection of Multiple Cracked Rotor Bars within an Induction Motor Using Both Vibration and Motor Current Analysis	263
H.	Detection of Electrical Problems within a Synchronous Motor Serving a Reciprocating Compressor Due to Loose Coils	269
XII.	**BELT DRIVE PROBLEMS**	
A.	Significant Improvement in Motor and Blower Vibration Due to Belt Replacement	276
B.	Effect of Tightening Loose Belts on Motor Vibration	280

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DETECTION AND CORRECTION OF FAULTS ON AN AIR HANDLER HAVING PROBLEMS DUE TO UNBALANCE, SHEAVE MISALIGNMENT, DEFECTIVE BELTS, RESONANCE AND A CRACKED FRAME SUPPORTING THE OUTBOARD FAN PILLOWBLOCK BEARING

I. BACKGROUND

The assistance of Technical Associates was requested to help resolve continuing failure problems on the belt driven air handler driven by a 200 HP motor pictured in Figure 1. Here, the motor operated at a speed of 1185 RPM while the fan speed was approximately 427 RPM. For several years, this fan had experienced failures, the most dramatic of which had been three instances where the fan shaft actually broke in half. In addition, both the motor and fan had experienced continuing bearing and belt failure problems.

Upon arrival at the plant site, Technical Associates inspected some of the failed components of this machine including two of the broken shafts. Close inspection of the failed shafts did not show a distinct fatigue failure pattern either due to bending or shear. However, the races particularly of the fan bearings tended to indicate misalignment with most of the loading concentrated on one side of the race. Following below will be a summary of the results found during the analysis, recommendations made and a look at the improvement in vibration spectra before and after corrective actions were taken.

II. RESULTS AND CONCLUSIONS

Vibration measurements were taken on each of the motor and fan bearing housings pictured in Figure 1. Plot 1 taken on the outboard motor bearing in the axial direction (Position 1A in Figure 1) showed high axial vibration with an overall level of .632 in/sec. Surprisingly, the motor running speed vibration at about 1192 CPM was only .090 in/sec. However, the dominant vibration on the motor was at 427 CPM with an excessive magnitude of .585 in/sec. Of course, the source of the 427 CPM vibration was the operating speed of the fan. Thus, this was a classic case often indicating sheave misalignment which will often generate excessive vibration at fan running speed in the axial direction when making measurements on the motor.

Next, another surprising spectrum is shown in Plot 2 which was taken in the axial direction on the sheave-side fan bearing at Position 3A (see Figure 1). Here, the vibration at 427 fan speed was low, but there was a very pronounced frequency at only 300 CPM dominating the vibration. Since this was suspected to be a frequency generated by belt speed (or multiple), the client was requested to review the purchase order for the motor and fan sheaves to confirm their pitch diameters as well as confirming the length of the belts. After this information was given, the belt speed (RPM) was calculated to be about 149.5 RPM (see Note 1). Therefore, this revealed the source of the 300 CPM vibration to be 2X belt defect frequency. It is not unusual for defects on belts to first generate a frequency at 2X belt speed (since the belt contacts two sheaves per revolution). However, what was unusual was the significant amplitude that resulted here at Position 3A in Plot 2. Therefore, the client was requested to stop the air handler to allow performance of impulse natural frequency tests. Such natural frequency testing was performed on each of the motor and fan bearing

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housings in horizontal, vertical and axial directions. Not surprisingly, when the test was performed on the sheave-side bearing in the axial direction, a pronounced axial natural frequency was found at 315 CPM as shown in Plot 3 (or only 5% away from the 300 CPM peak caused by 2X belt defect frequency). Therefore, even when defects would occur in the belts, their vibration would be greatly amplified by as much as 10 to 30 times higher due to resonance. And, if the belts suffered significant wear, resultant axial amplitudes would likely increase to highly unacceptable levels.

Next, the spectrum shown in Plot 4 indicated unbalance of the fan wheel as evidenced by a high amplitude .427 in/sec vibration at 427 RPM fan speed with the data taken on the same sheave-side bearing, but at Position 3H in the horizontal direction. Interestingly, note that none of the vibration at 2X belt defect frequency appeared whatsoever in the spectrum taken in this direction - another indication of the significant resonance problem this fan had in its axial direction. Phase analysis data was captured to ascertain the cause of the high 1X RPM fan vibration. This phase data is shown on Figure 1. Looking at the data taken on Positions 3 and 4, note that the phase difference in the horizontal direction between the outboard and inboard fan bearings (Positions 3 and 4) was approximately 110° while the vertical phase difference was approximately 130° between these two bearings. Since the phase difference was approximately the same in both the horizontal and vertical directions, and due to the high vibration 1X RPM, fan wheel unbalance was suspected.

NOTES:

1. Belt Speed = $\frac{\pi \times \text{Pulley RPM} \times \text{Pulley Pitch Diameter}}{\text{Belt Length}}$

(If Fan Sheave Pitch Diameter is entered, then Fan Sheave RPM must be also; and vice versa for the motor sheave.)

III. RECOMMENDATIONS

A. It was recommended that the belts be replaced and a close inspection be made of both the motor and fan sheaves to confirm they were in good condition. After replacing the belts (and sheaves if necessary), it was recommended to precision align the motor and fan sheaves with respect to one another.

B. It was also recommended that the frame of the fan be stiffened in the axial direction to raise its natural frequency above 315 CPM which would come very close to 2X belt speed when future belt defects did appear. However, it was cautioned that the natural frequency should not be raised to next coincide with either 3X or 4X belt speed. In other words, when stiffening attempts were made, an FFT analyzer was recommended to be on hand to ensure to what frequencies the frame natural frequency would be shifted.

C. After completion of recommendations A and B, it was recommended that a fan wheel balance should be performed in place to significantly lower the vibration at 1X fan speed below the .427 in/sec measured on May 18.

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IV. RESULTS OF CORRECTIVE ACTIONS

Corrective actions were attempted by Technical Associates on the same test day (May 18) as well as after this test. First, a hydraulic jack was placed in the axial direction on the fan frame in an attempt to temporarily shift the axial natural frequency away from 2X belt speed (and below 3X belt speed). However, the frame itself was of insufficient cross section to provide the necessary restraint to noticeably shift the natural frequency. Therefore, it was recommended on May 18 that the center distance between the motor and fan sheaves be changed which would not shift the frame stiffness or natural frequency, but would move the belt speed away from coincidence with the axial natural frequency.

Follow-up tests were performed on June 22 after the belts had been replaced and the motor had been moved further away from the fan, thereby increasing the center distance. This increase in center distance lowered the belt speed from about 150 RPM down to approximately 125 RPM. Therefore, now neither 2X belt speed (250 CPM) nor 3X belt speed (375 CPM) would be close to the fan frame axial resonance of about 315 CPM. Next, a fan wheel balance was performed in-place by Technical Associates on June 22. Plots 5 and 6 show the results of this balancing exercise on the inboard (Pos. 3H) and outboard (Pos. 4H) fan bearing. Even though the fan wheel showed noticeable wobble when it was slow-rolled, Plot 5 taken on Position 3H showed the field balance lowered vibration at fan speed from .444 in/sec down to .052 in/sec; while Plot 6 taken on Position 4H showed that it lowered the 1X RPM level from .365 in/sec down to only .052 in/sec.

However, please note the peak at 2X fan running speed in Plot 6 taken on Position 4H on the outboard fan bearing. Note that its level did not decrease (in fact, slightly increased to .241 in/sec) after the balance exercise. An examination of the frame supporting the outboard fan pillowblock bearing revealed a crack in the frame itself. This was pointed out to the client personnel who were advised to either replace and/or weld this frame.

V. LATER FOLLOW-UP AFTER WELDING OF THE OUTBOARD FAN PILLOWBLOCK FRAME

Plot 7 shows the spectrum taken after the frame supporting the outboard pillowblock fan bearing was welded. It lowered the vibration at 2X RPM from .241 in/sec down to .078 in/sec. Note that this spectrum was taken some time after the original corrective actions and that the vibration at 1X fan speed had now increased somewhat up to .193 in/sec. A check with the client showed that this was likely to buildup on the fan wheel. Still, a level less than .20 in/sec at 1X RPM on this fan wheel was quite acceptable and the noticeable drop in 2X RPM vibration was beneficial.

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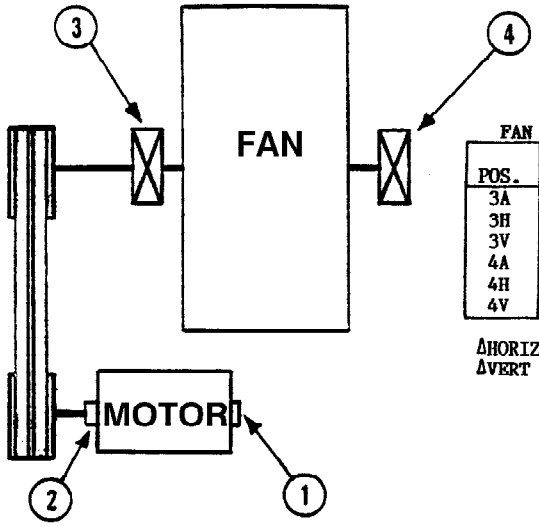
UNBALANCE DETECTION & CORRECTION ON A BELT-DRIVEN BLOWER

MACHINERY DATA GUIDE SHEET

RELIABILITY SURVEY — predictive maintenance program — (Page__of__)



MOTOR SPEED PHASE ANALYSIS

POS.	DIRECT PHASE	CORRECTED PHASE
1A	140°	140°
1H	210°	210°
1V	140°	140°
2A	305°	125°
2H	170°	170°
2V	220°	220°

FAN SPEED PHASE ANALYSIS

POS.	DIRECT PHASE	CORRECTED PHASE
3A	260°	260°
3H	300°	300°
3V	340°	340°
4A	100°	280°
4H	190°	190°
4V	250°	250°

$\Delta HORIZ_{3-4}$ = 300° − 190° = 110°
$\Delta VERT_{3-4}$ = 340° − 250° = 90°

MACHINE IDENT: L-12 AIR HANDLER

AREA:_____

ROUTE:_____

DRIVER	INTERMEDIATE	DRIVEN
RPM min : 1185	RPM in :	RPM min : 427.5
max :	out :	max :
MFGR :	RATIO:	MFGR. :
MODEL :	MODEL :	MODEL :
SER NO. :	SER NO. :	SER NO. :
V___ P___ A___	RATING :	RATING :
SHEAVE : ___DIA.	BELT # :	#BLADES :
BRG#: ___(I)	BRG#: ___(I)	BRG#: NU322 (I)
BRG#: ___(O)	BRG# ___(O)	BRG#: 6320 (O)
H.P.: 200	GEAR T: ___CT.	GEAR T: ___CT.
OTHER:	OTHER:	OTHER:

R-112492-1

FIGURE 1

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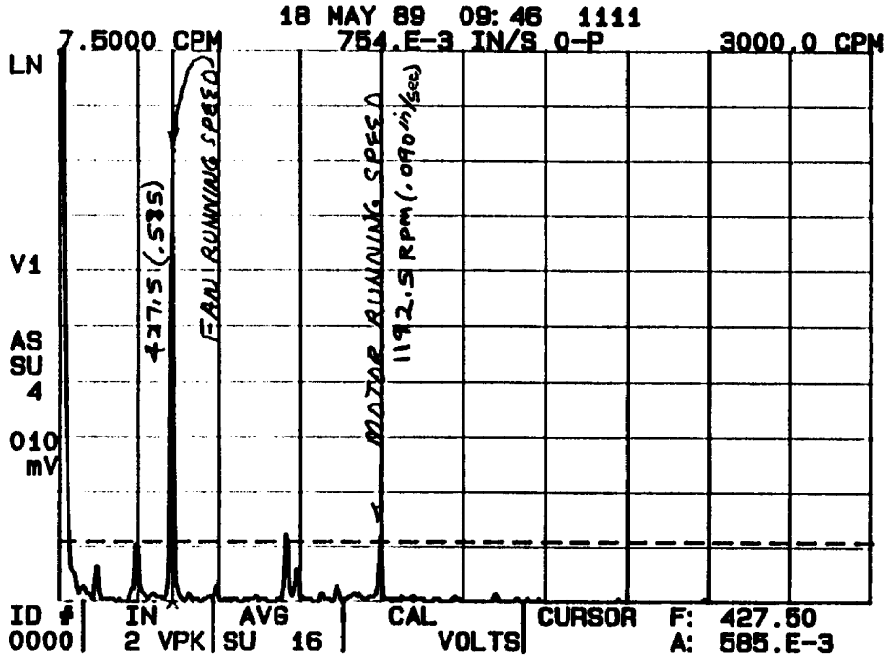

IN/S 0-P	CPM
585.E-3	427.50
92.9E-3	847.50
89.5E-3	1192.5

PEAK PICK ON TRACE 1

OVERALL = .632 in/sec

24-MAY-89

13: 17: 04

MOTOR VIBRATION
AXIALLY ON OUTBOARD END
POSITION 1A

PLOT 1

L-12AH

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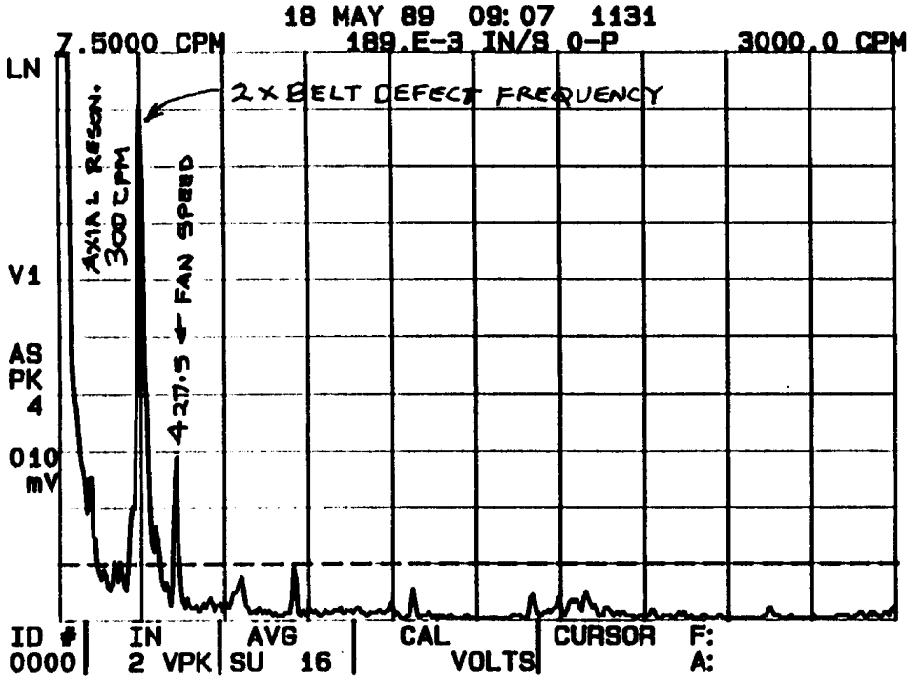

PEAK PICK ON TRACE 1

IN/S 0-P	CPM
48.3E-3	112.5
19.8E-3	202.4
19.8E-3	225.0
38.8E-3	270.0
171.E-3	300.0
31.9E-3	352.5
55.2E-3	427.5

24-MAY-89

12: 52: 05

INBOARD FAN BEARING
AXIAL - PEAK HOLD
POSITION 3A
PLOT 2

L-12AH

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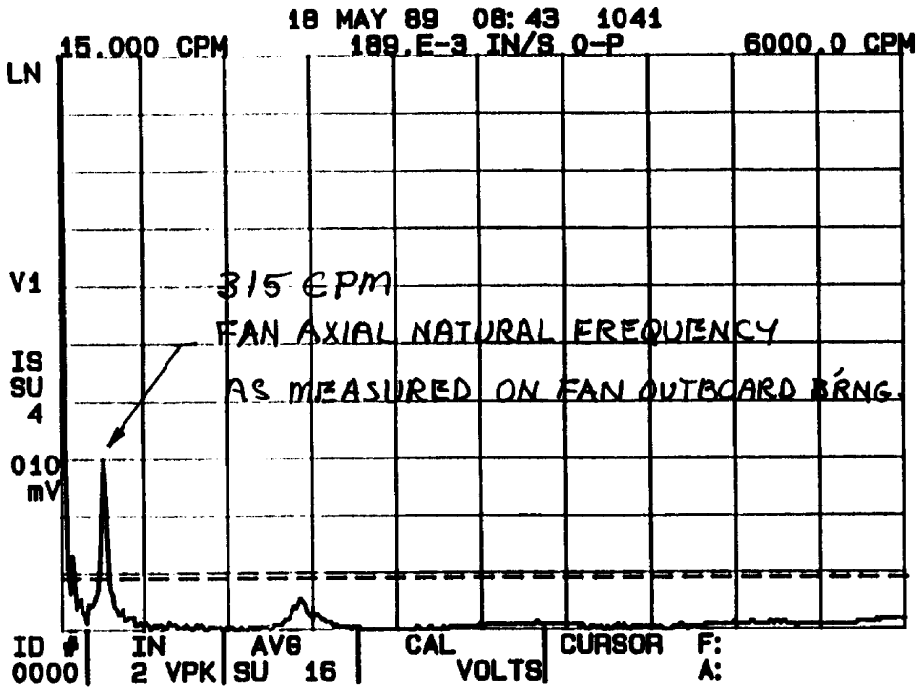

PEAK PICK ON TRACE 1

IN/S 0-P	CPM
25.0E-3	90.000
57.0E-3	315.00

24-MAY-89

12: 36: 05

IMPULSE TEST ON FAN OUTBOARD BEARING
POSITION 3A - IMPULSE IN AXIAL DIRECTION
PLOT 3

L-12AH

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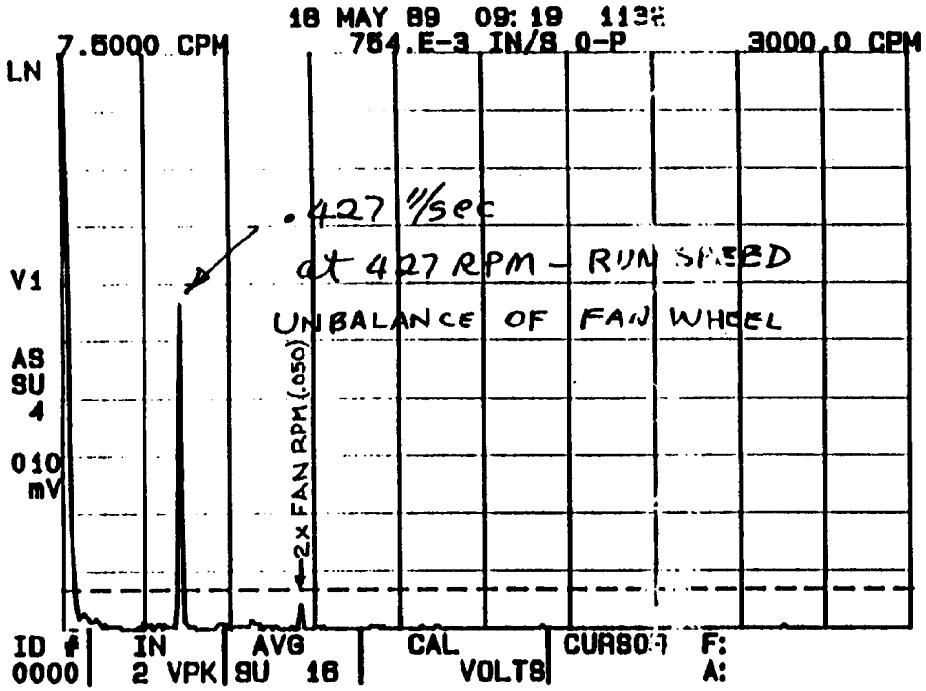


PEAK PICK ON TRACE 1

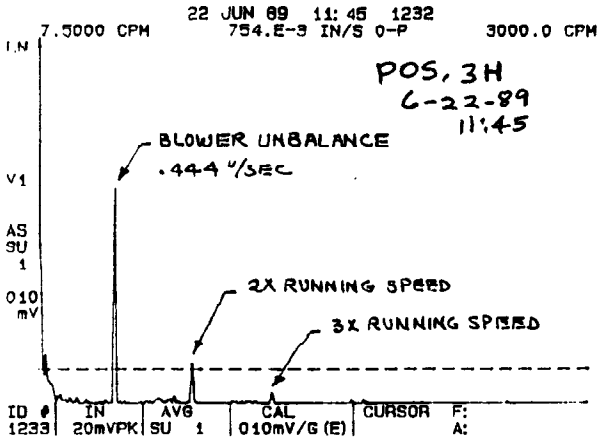
IN/S 0-P	CPM
427.E-3	427.50

24-MAY-89

12:55:56

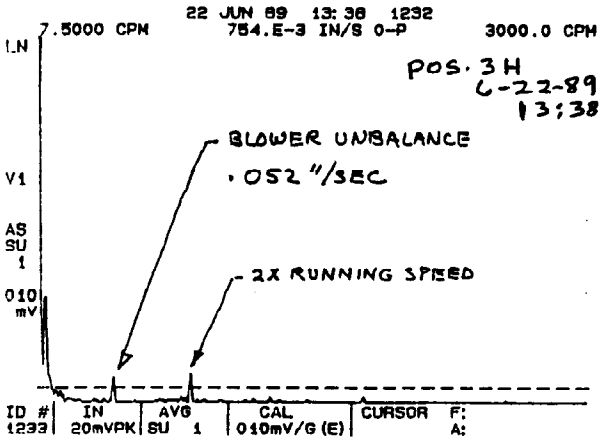
INBOARD FAN BEARING - HORIZONTAL
POSITION 3H
PLOT 4

L-12 AH



IN/S 0-P	CPM
99.8E-3	45.000
444.E-3	427.50
82.6E-3	855.00

BEFORE BALANCING



IN/S 0-P	CPM
217.E-3	52.500
51.6E-3	427.50
58.5E-3	855.00

AFTER BALANCING

22-JUN-89

15: 34: 00

UPPER PLOT = BEFORE BALANCING
.444"/sec

L-12 FAN

LOWER PLOT = AFTER BALANCING
.052"/sec

PLOT 5

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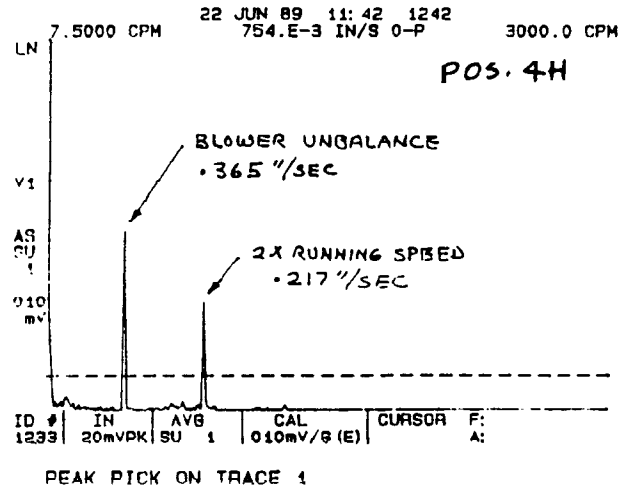
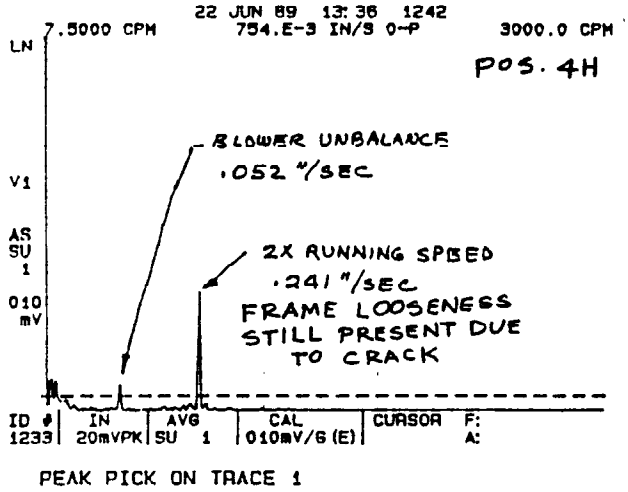


10

IN/S 0-P	CPM
365.E-3	427.50
217.E-3	855.00

BEFORE BALANCING



IN/S 0-P	CPM
65.4E-3	60.000
62.0E-3	82.500
51.6E-3	427.50
241.E-3	855.00

AFTER BALANCING

22-JUN-89

15: 31: 17

UPPER PLOT = BEFORE BALANCING
.365"/sec

L-12 FAN

LOWER PLOT = AFTER BALANCING
.052"/sec

PLOT 6

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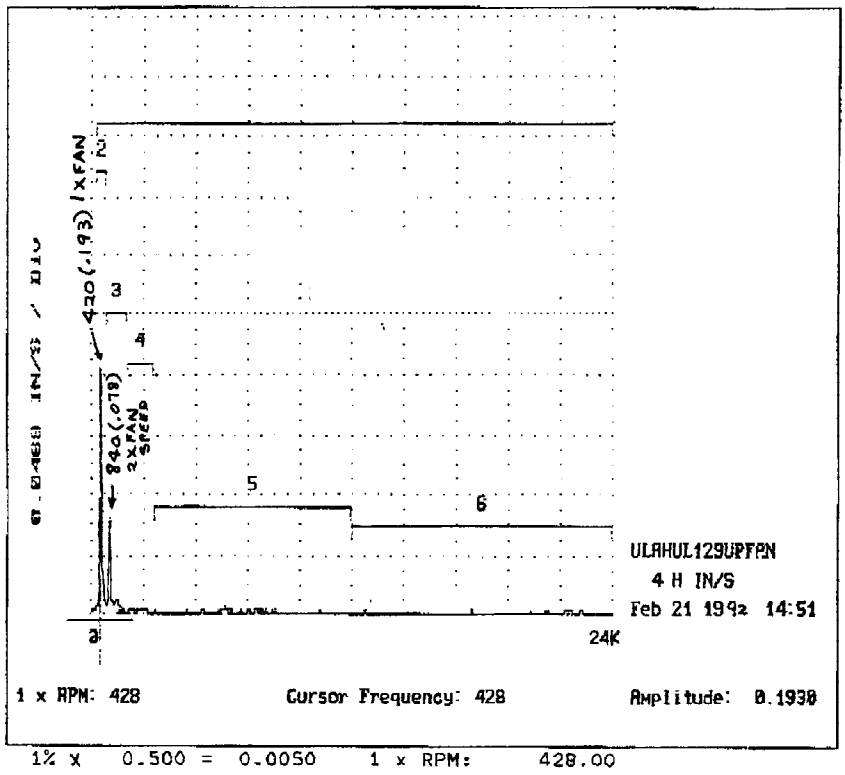

1% x 0.500 = 0.0050 1 x RPM: 428.00

Machine	Pos	Dir	Units	Date
ULAHUL12SUPFAN	4	H	IN/S	Feb 21 1980 14:51

Frequency	1 x RPM	Amplitude
420	0.98	0.1930
840	1.96	0.0775
1260	2.94	0.0125
6180	14.44	0.0060

T - Threshold adjust Q - Quit

AFTER WELDING FRAME SUPPORTING OUTBOARD FAN PILLOWBLOCK BEARING

PLOT 7

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DETECTION & CORRECTION OF UNBALANCE ON AN AIR CONDITIONER SUPPLY FAN

I. BACKGROUND

An HVAC contractor had been having vibration and associated noise difficulties with one of their customer's HVAC circulating fans. The contractor had changed bearings, adjusted belts, etc., without any success in improving vibration or noise levels and requested that Technical Associates perform a diagnostic investigation to determine the problem source.

II. RESULTS

The unit was belt driven as shown in Figure 1 with one (1) center mounted squirrel cage fan and one (1) overhung squirrel cage fan. The motor rotated at 1738 RPM and the fan at 738 RPM. All the spectra collected were dominated by a 720 CPM vibration with the highest amplitudes (axial and horizontal) at Position 4 on the fan shaft. It was determined that fan imbalance was the problem, and since Position 4 had the highest amplitude, balancing the overhung fan would produce the best results. What followed was a successful single plane balance with measurements from the Position 4 bearing. The weights were added at the inboard end of the overhung fan wheel on its backing plate close to bearing 4. Vibration levels at Position 4 fell from 1.74 in/sec at 720 CPM (fan rotational speed) to .035 in/sec at 720 CPM after the balance as seen by the "Before" and "After" spectra of Figure 2. The axial vibration at Position 4 also improved from 1.21 in/sec "Before" the balance to .448 in/sec "After" as seen in the Figure 3 spectra. The remaining axial response was attributed to:

1. Inadequate axial support
2. The variable pitch motor sheave
3. Sheave misalignment.

Noise levels also fell with the improved vibration.

III. CONCLUSIONS

1. The dominant problem was fan imbalance (168 grams of correction weights added to the outboard fan reduced unbalance to acceptable levels).
2. The axial response that remained after the balance was likely a result of:
   a. Structural flexing at the bearing supports
   b. The variable pitch motor sheave
   c. Sheave/belt misalignment.
3. Noise levels improved along with the vibration.
4. The simple single plane balance procedure worked well in correcting a significant imbalance in a complicated fan arrangement.

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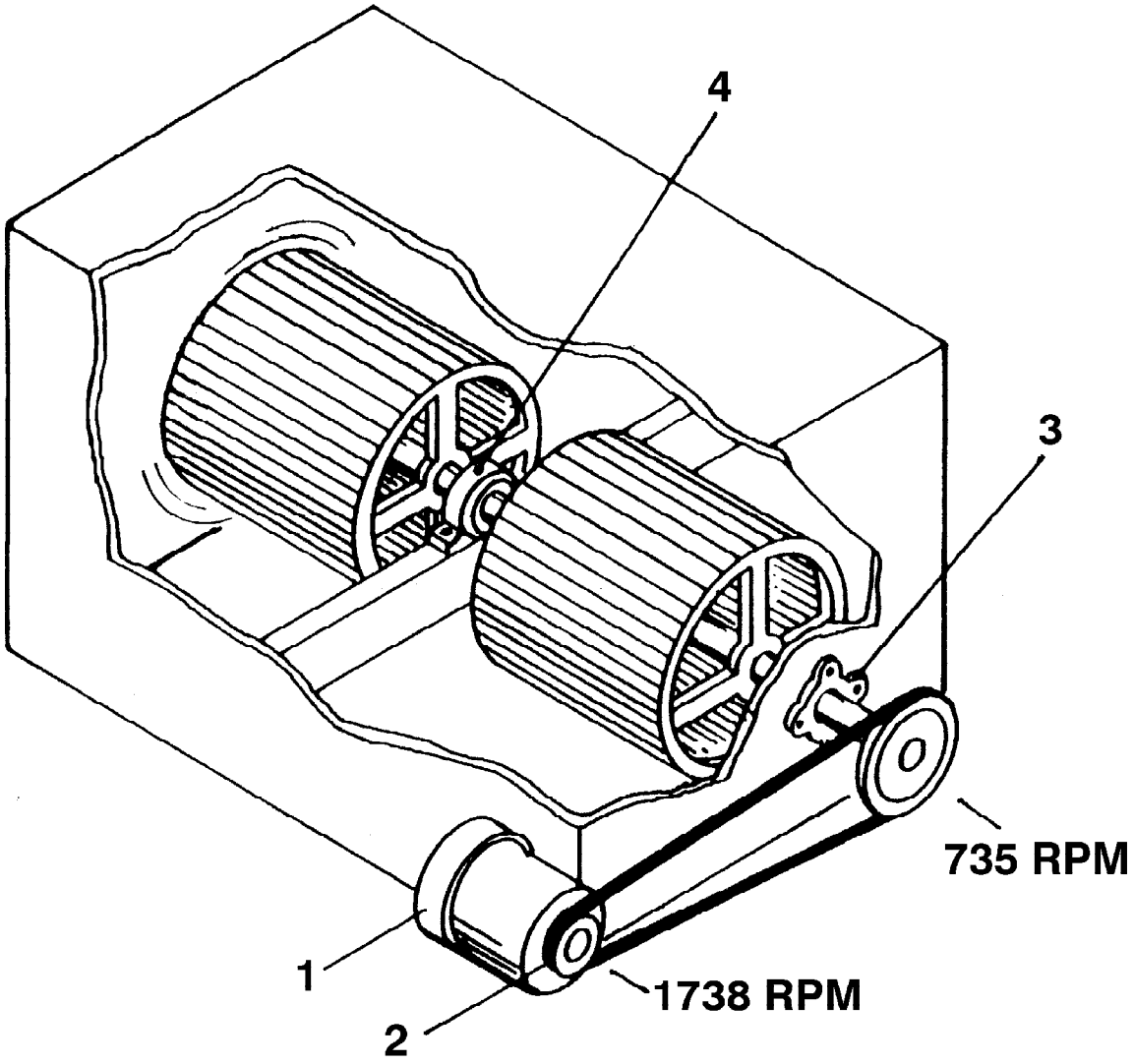
IV. RECOMMENDATIONS

1. Replace the motor's variable pitch sheave with a fixed diameter sheave of the proper size.

2. Align and tension the belts to specification.



SQUIRREL CAGE FAN ARRANGEMENT
FIGURE 1

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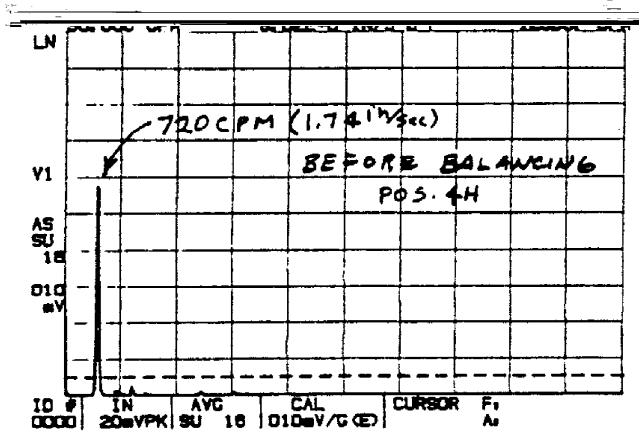


IN/S 0-P	CPM
1.74	720.00

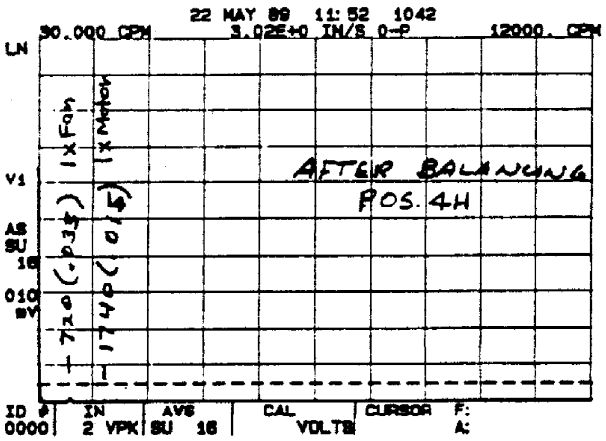
PEAK PICK ON TRACE 1

OA = 2.03

POSITION 4H

23-MAY-89

16: 33: 40



IN/S 0-P	CPM

PEAK PICK ON TRACE 1

PRECISION BALANCED

24-MAY-89

09: 26: 19

POSITION 4

HORIZONTAL "BEFORE" AND "AFTER" BALANCE SPECTRA

FIGURE 2

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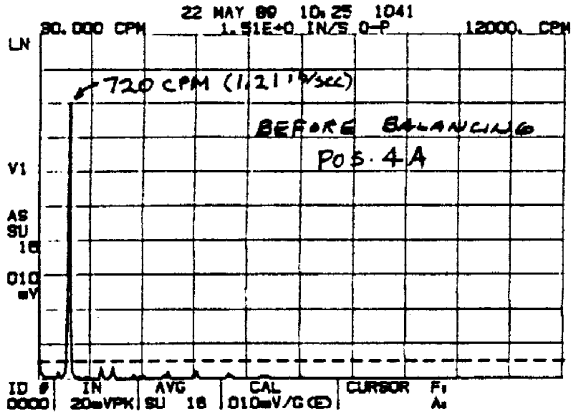


IN/S 0-P	CPM
1.21	720.00

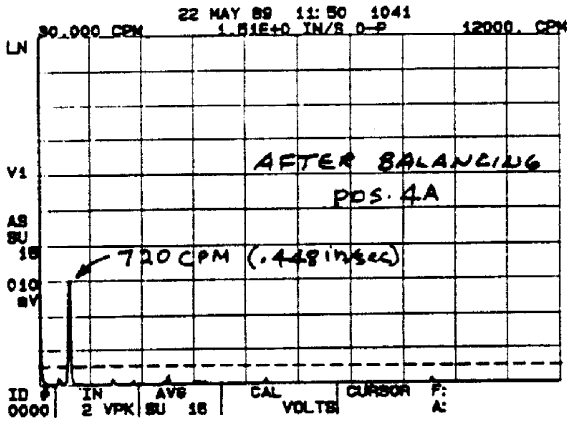
PEAK PICK ON TRACE 1

OA = 1.42

POSITION 4A

23-MAY-89

16: 32: 14



IN/S 0-P	CPM
448.E-3	720.00

PEAK PICK ON TRACE 1

1 INADEQUATE AXIAL SUPPORT BY FRAME (Esp. at IB Fan Brg.)
2 VARIABLE PITCH MOTOR SHEAVE IS PROBLEM
3 SHEAVES HAVE SOME MISALIGNMENT

24-MAY-89

RECOMMEND FIXED SHEAVE

09: 25: 34

POSITION 4
AXIAL "BEFORE" AND "AFTER" BALANCE SPECTRA
FIGURE 3

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DETECTION OF ECCENTRIC ORBIT ON A TURBINE/ GENERATOR DUE TO EXCESSIVE SLEEVE BEARING WEAR

I. BACKGROUND

Technical Associates was contacted by a client and requested to perform vibration diagnostic testing ASAP on their Turbine/Generator #3 which was triggering the alarm on its permanent monitoring system. The testing was accomplished later that night and in the early morning hours of the following day. Figure 11.11-G shows the arrangement of this turbine/generator with measurement positions. The machine was a 7500 KW unit running at 3600 RPM with a ten stage turbine. A Wavetek (Rockland) Model 5815A Real Time Analyzer (RTA) was used with a PCB 302A accelerometer. Data was also input to the RTA directly from the permanent monitoring system's transducers. An IRD 880 Swept Filter Analyzer (with strobe light) was also used for collecting phase data.

II. CONCLUSIONS/RECOMMENDATIONS

Spectral vibration data was collected in the axial, horizontal and vertical directions at each bearing housing using the PCB 302 accelerometer and the permanent monitoring system transducers. Phase readings were also made in each direction at each bearing housing. Table 11.11-B presents Technical Associates' conclusions based on the data collected on Turbine/Generator #3. An evaluation of Turbine/Generator #4 (a sister unit) was likewise included for contrast and comparison. The source of Turbine/Generator #3's excessive vibration levels was diagnosed as excessive bearing wear on the inboard exciter bearing, resulting in an eccentric orbit. Table 11.11-C is a table comparing overall peak velocity levels of Turbine/Generator #3 to two other Turbine/Generators' vibration levels at the same plant. This table made the excessive vibration amplitude at the inboard exciter bearing very obvious. Notice that the vibration amplitude is excessive in only the horizontal direction. Table 11.11-D presents the same machine-to-machine vibration level comparison with levels expressed in displacement (mils, Peak-Peak). Either way (velocity, or displacement) the inboard exciter bearing horizontal vibration levels were certainly excessive. Notice that in both cases the horizontal amplitude was approximately 4 times the vertical amplitude. Table 11.11-E presents the phase data comparison for the three Turbine/Generators in the plant. Notice that the horizontal and vertical phase readings for the inboard exciter bearing on Turbine/Generator #3 were identical. This indicates eccentricity in the shaft's orbit (see **Illustrated Vibration Diagnostic Chart**, Table I in Section 1.0), which in this case is caused by excessive clearance in the exciter inboard bearing. Phase differences of 0° (or 180°) between the horizontal and vertical directions indicate highly directional motion which is often associated with an eccentric orbit. The vibration spectra produced at the inboard and outboard exciter bearings in the horizontal direction is shown in Figure 11.11-H (Positions 4H & 5H - see Figure 11.11-G). These plots indicated a very high 1X RPM amplitude at the inboard exciter bearing, with no significant multiples of this frequency. However, the amplitude shown on the outboard bearing was relatively low (.698 in/sec at 4H versus only .077 in/sec at 5H; and compared with only .168 in/sec at 1X RPM on 4V).

The misalignment indicated by the axial spectra from either side of the generator/exciter coupling (see Table 11.11-B) was not confirmed by the phase reading and was probably a result of the excessive shaft movement at the inboard exciter bearing. Immediate exciter inboard bearing repair was recommended.

III. RESULTS

The entire exciter was replaced with a solid state design, not coupled to the turbine generator.

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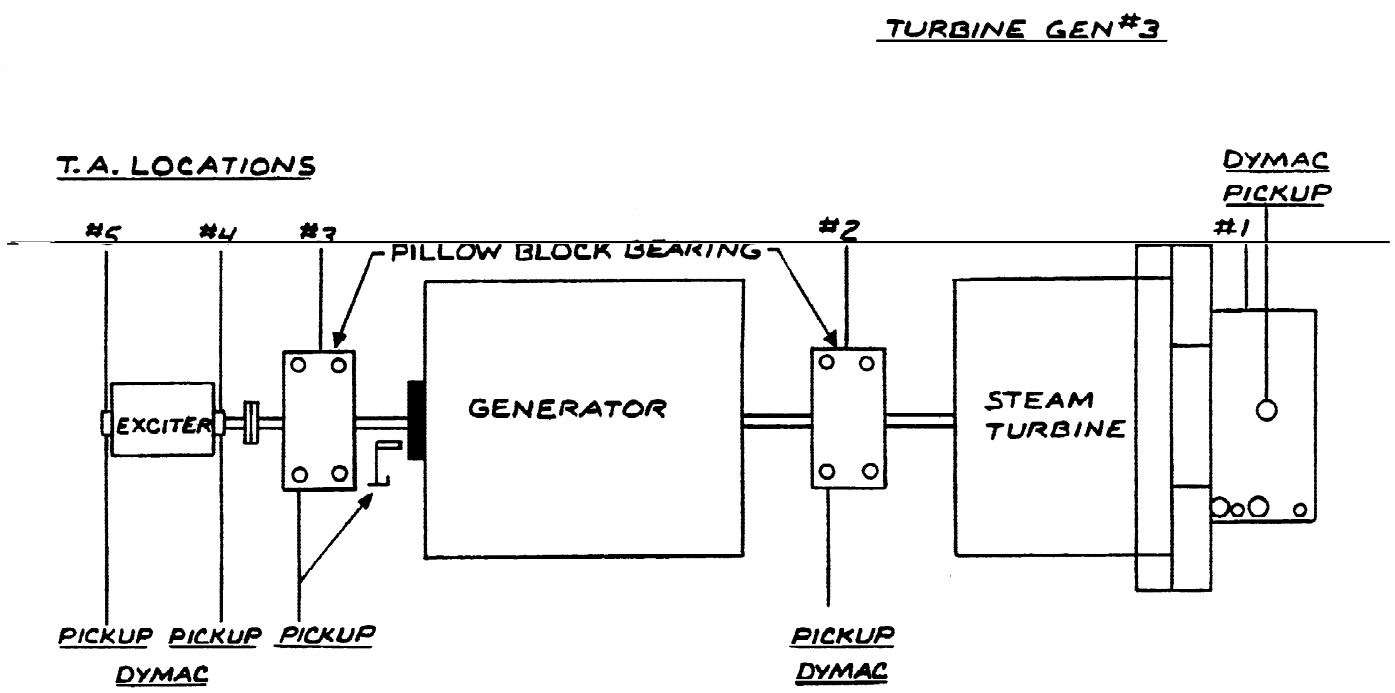

FIGURE 11.11-G

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CURRENT INDICATED CONDITION OF TURBINE/GENERATORS #3 & #4 AS OF 3/3/87

COMPONENT SYMPTOM	DIAGNOSTICS EVALUATION RESULTS	
	TURBINE/GENERATOR #3	TURBINE/GENERATOR #4
A. TURBINE		
1. Dynamic Balance	Good	Acceptable
2. Alignment/Coupling	Good	Slight Misalignment Indicated
3. Shaft Trueness	Good	Good
4. Bearing Condition	Good	Acceptable
5. Rotor/Stator Blading	Good	Good
6. Vibration Monitor	No Signal from OB Monitor	None on Machine
7. Flow	Slight Turbulence Indicated	Good
B. GENERATOR		
1. Dynamic Balance	Good	Acceptable
2. Alignment/Coupling	Misaligned with Exciter	Slight Misalignment Indicated
3. Shaft Trueness	Good	Good
4. Bearing Condition	Acceptable	Good
5. Electrical Problems	Good	Good
6. Vibration Monitor	Reading High (+90%)	None on Machine
C. EXCITER		
1. Dynamic Balance	Eccentric Orbit Indicated	Good
2. Alignment/Coupling	Misaligned with Generator	Symptoms indica., but very poss. slightly bent Exciter shaft
3. Shaft Trueness	Good	
4. Bearing Condition	IB Bearing Wear Indicated	Good
5. Electrical Problems	Acceptable	Good
6. Vibration Monitor	Both Read Slightly high (+40% to +50% high)	None on Machine
D. EXTERNAL SOURCES	None Indicated	Slight Contribution Indicated from Joy Recip. Compressors

: IMMEDIATE ACTION STRONGLY ADVISED. SEE RECOMMENDATIONS.

TABLE 11.11-B

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COMPARISON OF OVERALL VELOCITY MEASUREMENTS ON TURBINE/ GENERATORS #3, #4 AND #5 (in/sec O-Peak)

BEARING HOUSING LOCATION	UNIT #3		UNIT #4		① UNIT #5	
	POS.#	VEL.	POS.#	VEL.	POS.#	VEL.
Turbine Thrust Bearing	1H	.047	1H	.088	1H	.039
	1V	.084	1V	.043	1V	.072
	1A	.076	1A	.151	1A	.029
Outboard Turbine Bearing	②		2H	.086	2H	.033
			2V	.076	2V	.045
			2A	.067	2A	.031
Inboard Turbine Bearing	2H	.036	3H	(.221)	3H	.045
	2V	.053	3V	.131	3V	.039
	2A	.064	3A	.120	3A	.032
Inboard Generator Bearing	Shared with		4H	.112	5H	.102
	IB Turb. Brg.		4V	.111	5V	.065
	See Appendix		4A	.098	5A	.062
Outboard Generator Bearing	3H	.065	5H	.137	6H	.048
	3V	.020	5V	.055	6V	.061
	3A	.140	5A	(.263)	6A	.076
Inboard Exciter Bearing	4H	(.702)	6H	.055	7H	—
	4V	.186	6V	.067	7V	—
	4A	.243	6A	.160	7A	—
Outboard Exciter Bearing	5H	.089	7H	.064	8H	—
	5V	.072	7V	.084	8V	—
	5A	.223	7A	.090	8A	—

NOTES:

① Unit #5 appears identical to Unit #4 except for slight difference in arrangement of components between the turbine and generator. Due to this difference, no data was taken on the outboard end of the gearbox on Unit #4 as was done on Unit #5 (see Figures 2 and 3). Unit #5 was tested on January 9, 1987 by Technical Associates.

② Unit #3 - Using vendor drawing, it could not be ascertained whether or not there were separate outboard thrust and plain bearings with some distance between them as in Units #4 and #5, although this is likely the case. Only one (1) Dymac accelerometer monitor transducer was in place at this location.

③ (oval) = Excessive vibration above "UNACCEPTABLE" severity limit. IMMEDIATE ACTION RECOMMENDED!

④ ( ) = High vibration within "ROUGH" severity specified in Table 1A Severity Table.

TABLE 11.11-C

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COMPARISON OF OVERALL VELOCITY MEASUREMENTS ON TURBINE/ GENERATORS #3, #4 AND #5 (mils Peak-Peak)

BEARING HOUSING LOCATION	UNIT #3		UNIT #4		① UNIT #5	
	POS.#	DISP.	POS.#	DISP.	POS.#	DISP.
Turbine Thrust Bearing	1H	.054	1H	.476	1H	.136
	1V	.273	1V	.158	1V	.213
	1A	.053	1A	.814	1A	.114
Outboard Turbine Bearing	②		2H	.331	2H	.095
			2V	.351	2V	.097
			2A	.113	2A	.085
Inboard Turbine Bearing	2H	.135	3H	.828	3H	.167
	2V	.258	3V	.574	3V	.096
	2A	.217	3A	.577	3A	.056
Inboard Generator Bearing	Shared with IB Turb. Brg.		4H	.470	5H	.290
			4V	.597	5V	.338
			4A	.513	5A	.249
Outboard Generator Bearing	3H	.326	5H	.741	6H	.137
	3V	.074	5V	.268	6V	.335
	3A	.588	5A	(1.263)	6A	.293
Inboard Exciter Bearing	4H	(3.804)	6H	.240	7H	—
	4V	.817	6V	.260	7V	—
	4A	1.254	6A	.736	7A	—
Outboard Exciter Bearing	5H	.431	7H	.347	8H	—
	5V	.324	7V	.434	8V	—
	5A	1.148	7A	.373	8A	—

NOTES:

① Unit #5 appears identical to Unit #4 except for slight difference in arrangement of components between the turbine and generator. Due to this difference, no data was taken on the outboard end of the gearbox on Unit #4 as was done on Unit #5 (see Figures 2 and 3). Unit #5 was tested on January 9, 1987 by Technical Associates.

② Unit #3 - Using vendor drawing, it could not be ascertained whether or not there were separate outboard thrust and plain bearings with some distance between them as in Units #4 and #5, although this is likely the case. Only one (1) Dymac accelerometer monitor transducer was in place at this location.

③ (oval) = Excessive vibration above "UNACCEPTABLE" severity limit. IMMEDIATE ACTION RECOMMENDED!

④ ( ) = High vibration within "ROUGH" severity specified in Table 1A Severity Table.

TABLE 11.11-D

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COMPARISON OF VIBRATION PHASE MEASUREMENTS ON TURBINE/ GENERATORS #3, #4 AND #5

BEARING HOUSING LOCATION	UNIT #3		UNIT #4		UNIT #5	
	POS.#	① PHASE	POS.#	① PHASE	POS.#	① PHASE
Turbine Thrust Bearing	1H	1:00	1H	9:00	1H	7:00
	1V	11:30	1V	3:00	1V	6:00
	1A	7:00	1A	3:00	1A	12:30
Outboard Turbine Bearing			2H	4:00	2H	6:30
			2V	9:00	2V	3:00
			2A	2:00	2A	6:00
Inboard Turbine Bearing	2H	3:00	3H	6:00	3H	1:30
	2V	1:00	3V	2:00	3V	11:00
	2A	7:00	3A	11:00	3A	4:30
Inboard Generator Bearing	Shared with IB Turb. Brg.		4H	9:00	5H	6:00
			4V	11:00	5V	6:30
			4A	11:00	5A	2:00
Outboard Generator Bearing	3H	1:00	5H	2:00	6H	6:00
	3V	3:00	5V	9:00	6V	12:30
	3A	7:00	5A	8:00	6A	1:00
Inboard Exciter Bearing	4H	5:00	6H	12:00	7H	---
	4V	5:00	6V	5:00	7V	---
	4A	4:00	6A	11:00	7A	---
Outboard Exciter Bearing	5H	3:00	7H	11:00	8H	---
	5V	12:00	7V	6:00	8V	---
	5A	6:00	7A	8:00	8A	---

NOTES:

① Phase measured by IRD 880 swept filter analyzer and IRD 970 accelerometer.

TABLE 11.11-E

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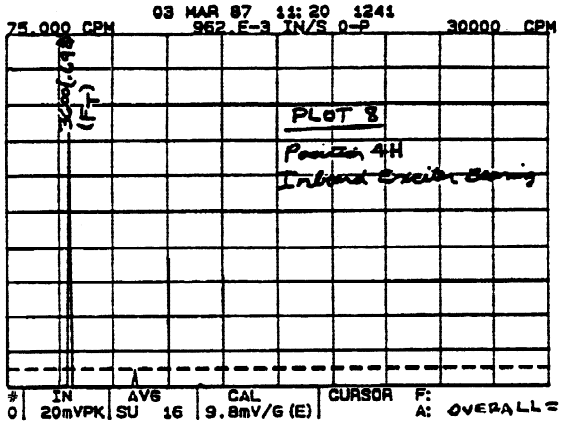


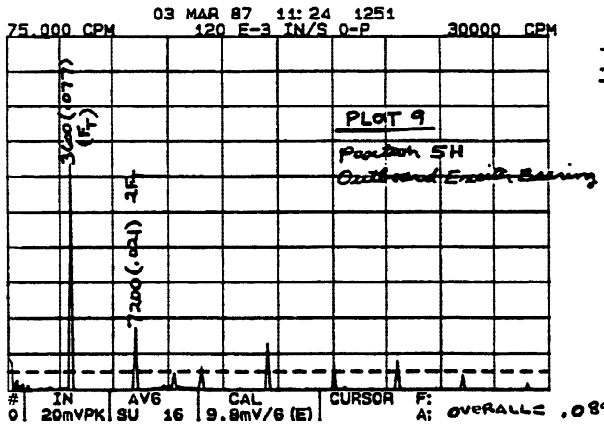
PEAK PICK ON TRACE 1

IN/S 0-P	CPM
698.E-3	3600.0

04-MAR-87

10: 46: 46

TURB/GEN 3



PEAK PICK ON TRACE 1

IN/S 0-P	CPM
76.7E-3	3600.0
20.8E-3	7200.0
7.67E-3	10800.
15.9E-3	14400.
8.77E-3	18000.
9.86E-3	21599.

04-MAR-87

10: 57: 39

TURB/GEN 3

FIGURE 11.11-H

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EFFECT OF COUPLING MISALIGNMENT ON VIBRATION OF A WATER SCRUBBER PUMP

I. BACKGROUND

During a July 13, 1989 predictive maintenance vibration survey by Technical Associates, a horizontally mounted centrifugal water pump (1-1 Scrubber Pump) was detected as having a sudden axial vibration increase. See Figure 1 for the machine arrangement. The increase is shown in the trend graphs and waterfall plots of Figures 2, 3 and 4 taken on Positions 1A and 2A as shown. Notice that the waterfall spectra show the dominant vibration was at 1X RPM (nominal 1800 RPM). Since the motor was having other problems, our client decided to replace the motor instead of just aligning it. A new coupling was unavailable at that time, so the old coupling was reused. The next survey of 9/7/89 had reduced axial measurements indicating the unit was well aligned as seen by the trends and waterfall plots of Figures 2 and 4. The axial vibration remained stable until the October 5, 1989 survey. This survey found the axial motor vibration had again increased to levels in excess of 1.0 in/sec as seen in Figures 2, 3 and 4. This time, the problem was found to be caused by excessive wear of the "used" coupling. After the coupling was replaced and the unit aligned, vibration levels again fell to acceptable levels as seen in the 11/9/89 measurements of Figure 5 and the trends and spectra of Figures 2 and 3.

II. CONCLUSIONS

A. 1-1 Scrubber Pump's first misalignment condition resulted from a motor problem and the second from a coupling problem.

B. Vibration measurements after the final alignment indicate the unit was well aligned.

III. RECOMMENDATIONS

A. Recommend continuing the periodic vibration measurements on a monthly basis.

IV. RESULTS OF CORRECTIVE ACTIONS

A. The final alignment with the new coupling has remained stable, at least through November, 1991 as seen in the Trend Graphs of Figures 6 and 7.

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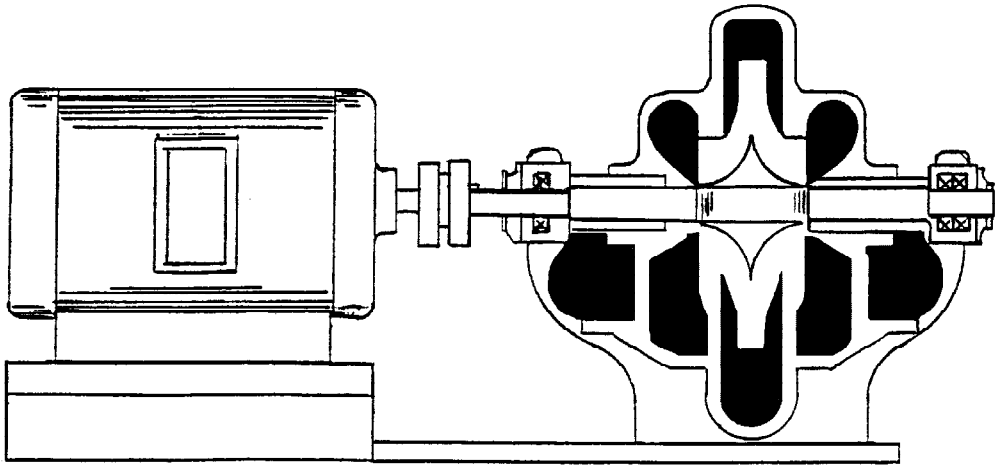
MACHINERY DATA GUIDE SHEET

RELIABILITY SURVEY predictive maintenance program (Page__of__)



MACHINE IDENT: 1-1 SCRUBBER PUMP

AREA:_____

ROUTE:_____

DRIVER	INTERMEDIATE	DRIVEN
RPM min :_____	RPM in : _____	RPM min :_____
max :1770	out : _____	max :_____
MFGR :TOSHIBA	RATIO:_____	MFGR. :WORTHINGTON
MODEL :B2004FKF4BD	MODEL :_____	MODEL :10LR15
SER NO. :4903316	SER NO. :_____	SER NO. :60-740830
V 460 P 3 A 230	RATING :_____	RATING :_____
SHEAVE :_____DIA.	BELT # :_____	#BLADES :_____
BRG#:_____(I)	BRG#:_____(I)	BRG#:_____(I)
BRG#:_____(0)	BRG#_____(0)	BRG#:_____(0)
H.P.: 200	GEAR T:_____CT.	GEAR T:_____CT.
OTHER:_____	OTHER:_____	OTHER:_____

FIGURE 1

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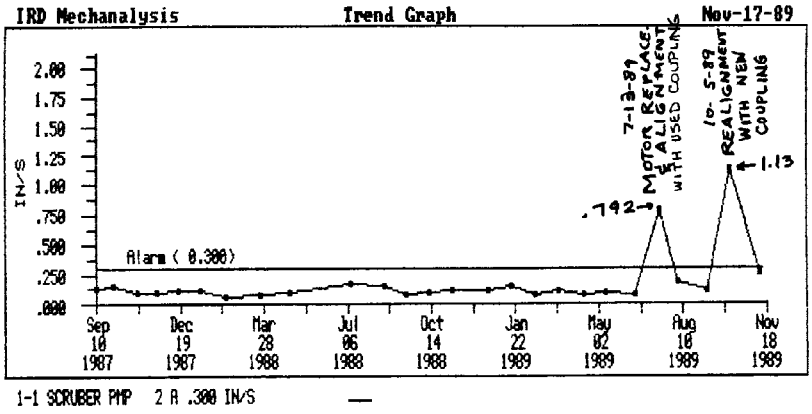

IRD Mechanalysis Data Analysis System

Machine 1-1 SCRUBER PMP Pos 2 Dir A Units IN/S Alarm .300

Ampl	Insp Code	Date mm/dd/yy	Time	Ampl	Insp Code	Date mm/dd/yy	Time
.134	0	09-10-87	0:00	.114	0	11-07-88	12:16
.159	0	09-30-87	0:00	.122	0	12-20-88	12:46
.099	0	10-28-87	0:00	.147	0	01-17-89	11:05
.099	0	11-20-87	0:00	.075	0	02-14-89	10:47
.114	0	12-16-87	0:00	.117	0	03-14-89	10:44
.114	0	01-11-88	0:00	.078	0	04-13-89	12:00
.066	0	02-11-88	0:00	.103	0	05-09-89	9:33
.084	0	03-23-88	0:00	.078	0	06-13-89	12:57
.094	0	04-27-88	0:00	.792	0	07-13-89	11:34
.164	0	07-09-88	0:00	.190	0	08-03-89	13:10
.155	0	08-18-88	12:43	.116	0	09-07-89	10:49
.085	0	09-13-88	15:12	1.13	0	10-05-89	10:51
.099	0	10-11-88	12:46	.258	0	11-09-89	12:20

FIGURE 2

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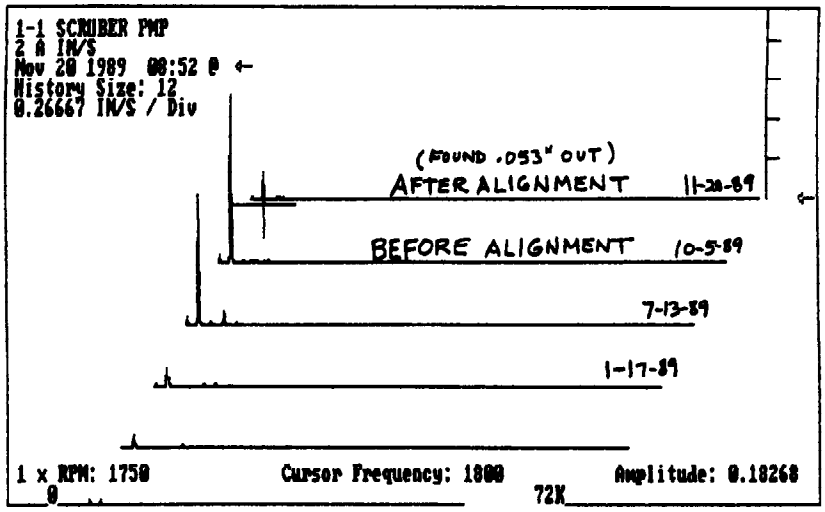

2% x 2.0000 = 0.04000 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
1-1 SCRUBER PMP	2	A	IN/S	Nov 20 1989 08:52

Frequency	1 x RPM	Amplitude
1800	1.03	0.18268

2% x 2.0000 = 0.04000 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
1-1 SCRUBER PMP	·2	A	IN/S	Oct 05 1989 10:51

Frequency	1 x RPM	Amplitude
180	0.10	0.04988
1800	1.03	1.09992

2% x 2.0000 = 0.04000 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
1-1 SCRUBER PMP	2	A	IN/S	Jul 13 1989 11:34

Frequency	1 x RPM	Amplitude
180	0.10	0.04988
1800	1.03	0.82986
5400	3.09	0.08483

2% x 2.0000 = 0.04000 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
1-1 SCRUBER PMP	2	A	IN/S	Jan 17 1989 11:05

Frequency	1 x RPM	Amplitude
1800	1.03	0.10898

FIGURE 3

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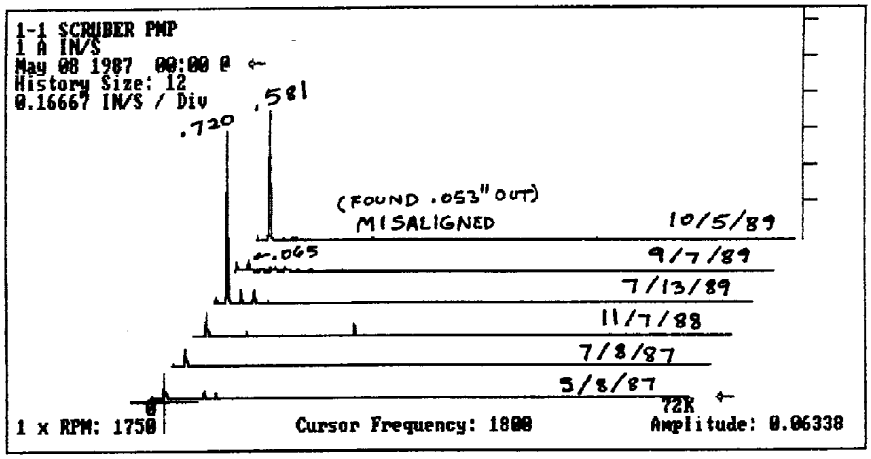
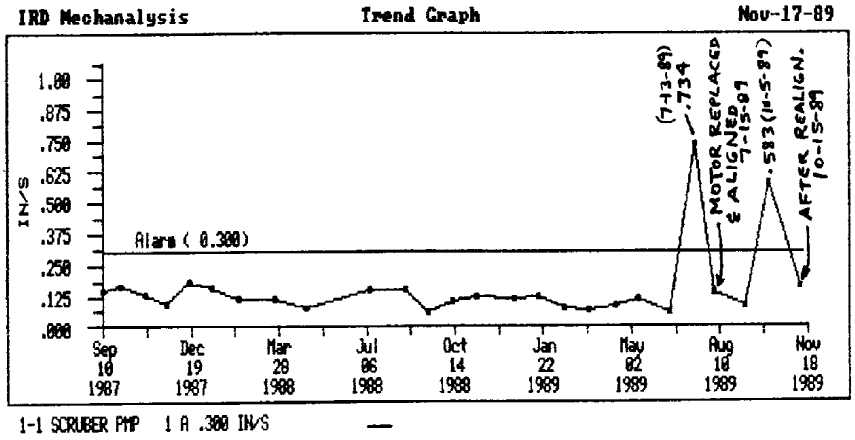
EFFECT ON VIBRATION OF ALIGNMENT





IRD Mechanalysis Data Analysis System

Machine 1-1 SCRUBER PMP Pos 1 Dir A Units IN/S Alarm .300

Ampl	Insp Code	Date mm/dd/yy	Time
.144	0	09-10-87	0:00
.164	0	09-30-87	0:00
.129	0	10-28-87	0:00
.094	0	11-20-87	0:00
.184	0	12-16-87	0:00
.159	0	01-11-88	0:00
.109	0	02-11-88	0:00
.109	0	03-23-88	0:00
.079	0	04-27-88	0:00
.144	0	07-09-88	0:00
.145	0	08-18-88	12:42
.060	0	09-13-88	15:11
.099	0	10-11-88	12:45
.124	0	11-07-88	12:15
.111	0	12-20-88	12:45
.117	0	01-17-89	11:04
.077	0	02-14-89	10:47
.068	1	03-14-89	10:43
.083	0	04-13-89	12:00
.114	0	05-09-89	9:33
.061	0	06-13-89	12:57
.734	0	07-13-89	11:33
.140	0	08-03-89	13:09
.085	0	09-07-89	10:49
.583	0	10-05-89	10:49
.171	0	11-09-89	12:19

FIGURE 4

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IRD Mechanalysis Data Analysis System Page 1
Nov-17-89

Machine Report (Latest Data) for: 1-1 SCRUBER FMP

Date	Time	Pos	Dir	Ampl	Units	Change	Alarm	Inspection
Nov-09-89	12:19	1	A	.171	IN/S	-71%	.300	
Nov-09-89	12:19	1	H	.079	IN/S	-64%	.300	
Nov-09-89	12:19	1	H	.262	g/SE	134%	.600	
Nov-09-89	12:19	1	V	.071	IN/S	-49%	.300	
Nov-09-89	12:20	2	A	.258	IN/S	-77%	.300	
Nov-09-89	12:20	2	H	.061	IN/S	-73%	.300	
Nov-09-89	12:20	2	H	.175	g/SE	-8%	.600	
Nov-09-89	12:20	2	V	.174	IN/S	-61%	.300	
Nov-09-89	12:20	3	A	.202	IN/S	2%	.300	
Nov-09-89	12:20	3	H	.116	IN/S	10%	.300	
Nov-09-89	12:20	3	H	.228	g/SE	24%	.600	
Nov-09-89	12:20	3	V	.156	IN/S	47%	.300	
Nov-09-89	12:21	4	A	.170	IN/S	-1%	.300	
Nov-09-89	12:21	4	H	.112	IN/S	-7%	.300	
Nov-09-89	12:21	4	H	.127	g/SE	7%	.600	
Nov-09-89	12:21	4	V	.118	IN/S	-2%	.300	

16 lines this report

Arrows Show Little Improvement in Spike Energy as Expected

FIGURE 5

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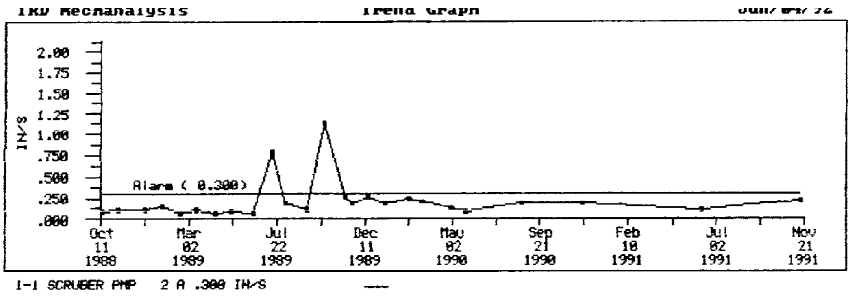

IRD Mechanalysis Data Analysis System

Machine 1-1 SCRUBER PMP Pos 2 Dir A Units IN/S Alarm .300

Ampl	Insp Code	Date mm/dd/yy	Time	Ampl	Insp Code	Date mm/dd/yy	Time
.099	0	10/11/88	12:46	.258	0	11/09/89	12:20
.114	0	11/07/88	12:16	.187	0	11/20/89	8:52
.122	0	12/20/88	12:46	.257	0	12/15/89	10:08
.147	0	01/17/89	11:05	.187	0	01/11/90	9:01
.075	0	02/14/89	10:47	.244	0	02/20/90	8:22
.117	0	03/14/89	10:44	.199	0	03/13/90	8:20
.078	0	04/13/89	12:00	.134	0	04/30/90	7:45
.103	0	05/09/89	9:33	.103	0	05/24/90	10:57
.079	0	06/13/89	12:57	.103	0	05/24/90	10:37
.792	0	07/13/89	11:34	.194	0	08/23/90	11:55
.190	0	08/03/89	13:10	.194	0	11/29/90	10:59
.116	0	09/07/89	10:49	.113	0	06/06/91	10:32
1.13	0	10/05/89	10:51	.228	0	11/14/91	9:04

FIGURE 7

FIGURE 6

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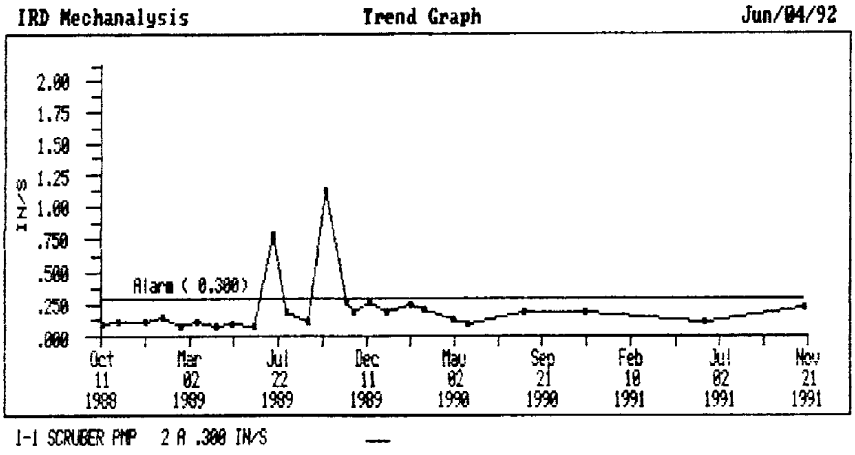

IRD Mechanalysis Data Analysis System

Machine 1-1 SCRUBER PMP Pos 2 Dir A Units IN/S Alarm .300

Ampl	Insp Code	Date mm/dd/yy	Time	Ampl	Insp Code	Date mm/dd/yy	Time
.099	0	10/11/88	12:46	.258	0	11/09/89	12:20
.114	0	11/07/88	12:16	.187	0	11/20/89	8:52
.122	0	12/20/88	12:46	.257	0	12/15/89	10:08
.147	0	01/17/89	11:05	.187	0	01/11/90	9:01
.075	0	02/14/89	10:47	.244	0	02/20/90	8:22
.117	0	03/14/89	10:44	.199	0	03/13/90	8:20
.078	0	04/13/89	12:00	.134	0	04/30/90	7:45
.103	0	05/09/89	9:33	.103	0	05/24/90	10:57
.078	0	06/13/89	12:57	.103	0	05/24/90	10:37
.792	0	07/13/89	11:34	.194	0	08/23/90	11:55
.190	0	08/03/89	13:10	.194	0	11/29/90	10:59
.116	0	09/07/89	10:49	.113	0	06/06/91	10:32
1.13	0	10/05/89	10:51	.228	0	11/14/91	9:04

FIGURE 7

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RESOLUTION OF A SEVERE RESONANCE PROBLEM AT 2X RPM WHICH HAD CAUSED CONTINUING IMPELLER SHAFT FRACTURES ON A CERTAIN MODEL OF VERTICAL PUMP

I. BACKGROUND

Figure 1 shows the design configuration for the subject vertical pump. This particular model of pump had been used at a number of municipal treatment stations and had experienced a large number of fatigue failures of the pump shaft at a cross section close to the impeller locking nut. Originally, vibration had been so high that the locking nuts had been backing off. As a result of this, a strong Loctite epoxy was applied to the locking nuts which successfully prevented them from backing off. However, then a number of pumps began experiencing the catastrophic fatigue failures of impeller shafts.

As a result, Technical Associates was contacted with the purpose of measuring vibration, evaluating its severity, diagnosing potential problems and making necessary recommendations to resolve any problems diagnosed.

II. ANALYSIS RESULTS

Figure 1 shows the locations at which measurements were made on both the motor and vertical pump. Looking at Figure 1, note that the system is made up of a motor which drives a hollow 3.5 inch shaft of approximately 74 inch length which then drives the pump at a speed of approximately 1780 RPM. Highest vibration was found on the pump, particularly at Positions 3V and 4V (see Figure 1). Plot 1 shows a sample spectrum taken on Position 3V which was actually made in the horizontal direction parallel to the floor. Note the excessive overall vibration level of .714 in/sec which was dominated by .650 in/sec at 3570 CPM (2X running speed), compared to only .181 in/sec at 1770 RPM operating speed. It should be pointed out that the impeller is outfitted with 2 vanes.

A number of tests were then conducted to determine why 2X RPM vibration dominates the spectrum. These included impulse natural frequency tests which were made in the motor, shaft and pump. Plot 2 shows the results of such an impulse test. Plot 2 was captured when the pump was impacted in the horizontal direction directly in line with positions 3V and 4V. It showed a predominant natural frequency at 3780 CPM which was only 5.9% away from the 3570 CPM vane pass frequency. Referring to Figure 1 and Plot 2, note that the 2000 CPM natural frequency was not involved since it was simply the resonance of the safety guard screen which was positioned around the pump and shaft directly above Positions 3H and 3V.

Because the proximity of the natural frequency was so close to vane pass frequency (2X RPM), it was recommended that a simple test be preformed to evaluate how much vibration could potentially be reduced if the natural frequency was moved away from the vane pass frequency. Figure 2 shows that temporary stiffeners were placed between the pump and walls in both the east and north directions of this pump. The stiffening materials were simply a 4" x 4" between the east wall and pump and two 2" x 4"s between the north wall and pump (see Figure 2).

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III. RESULTS OF CORRECTIVE ACTIONS

Plot 3 shows the results of the temporary stiffening, how high it moved the natural frequency and what were the resultant effects on running vibration. Note the Plot 3A is just a repeat of Plot 1 showing the vibration spectrum before stiffening. Plot 3B shows the spectrum after stiffening. Plot 3B shows that the natural frequency was moved up from 3780 CPM to 3960 CPM. Although this was only an increase of 4.8% higher, the vane pass vibration dropped from .650 in/sec down to .196 in/sec after stiffening with the temporary boards. Later, a more permanent design modification was made and installed which moved the natural frequency above 4200 CPM resulting in a final vane pass vibration of less than .12 in/sec. It is understood that no further impeller shaft failures have occurred as of this writing which is 37 months from the date of this testing.

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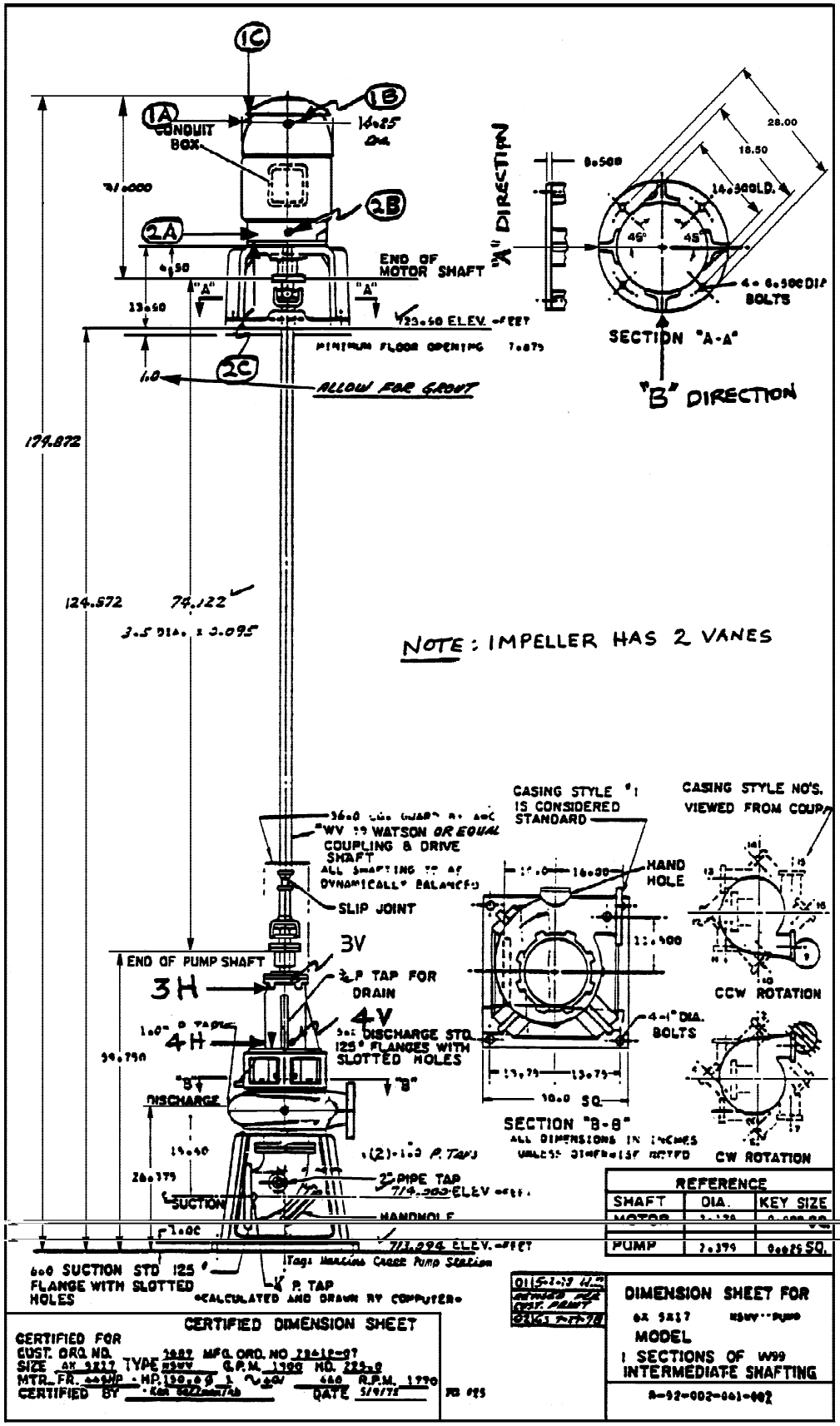

FIGURE 1. MEASUREMENT LOCATIONS ON THE MOTOR & VERTICAL PUMP

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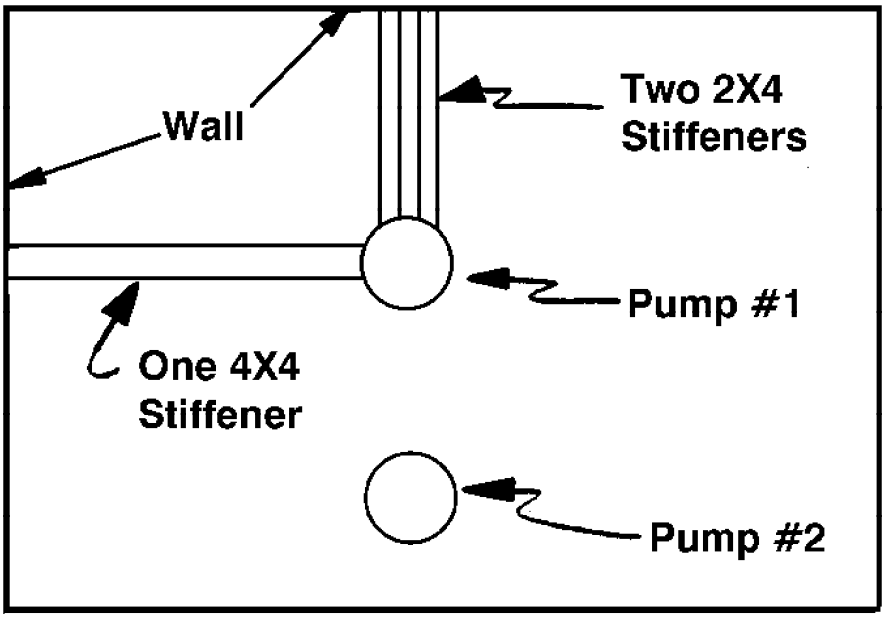

FIGURE 2.
INSTALLATION OF TEMPORARY STIFFENING BRACES TO INCREASE NATURAL FREQUENCY AWAY FROM VANE PASS FREQUENCY

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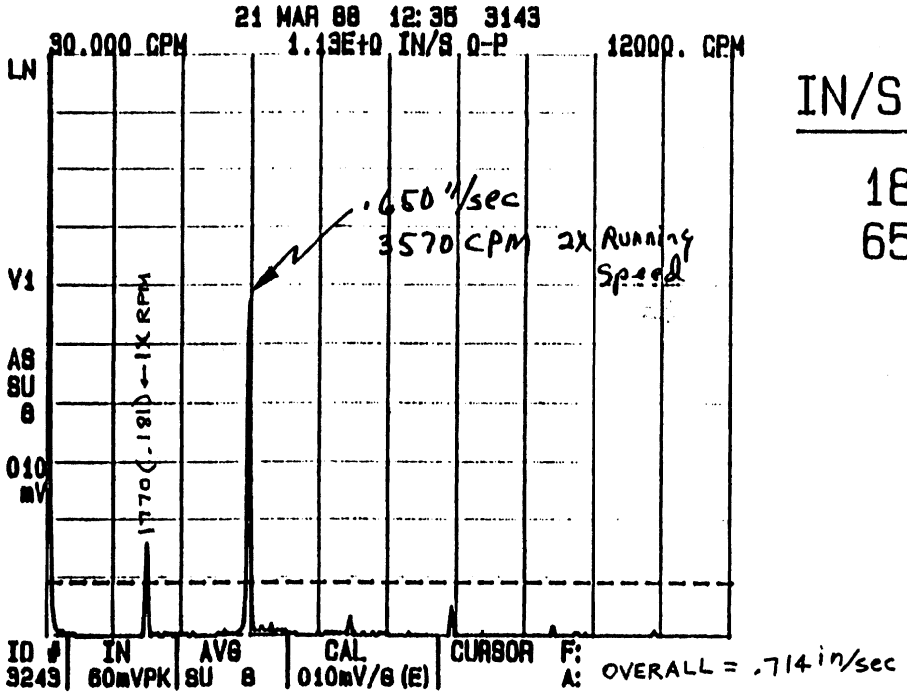


IN/S 0-P	CPM
181.E-3	1770.0
650.E-3	3570.0

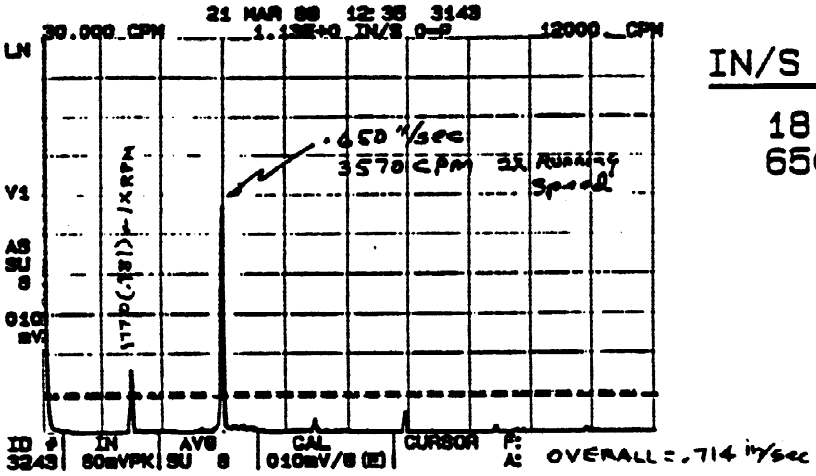
PEAK PICK ON TRACE 1

PLOT 1.

21-MAR-88

POSITION 4V
RUNNING VIBRATION

18: 53: 29



IN/S 0-P	CPM
181.E-3	1770.0
650.E-3	3570.0

PEAK PICK ON TRACE 1

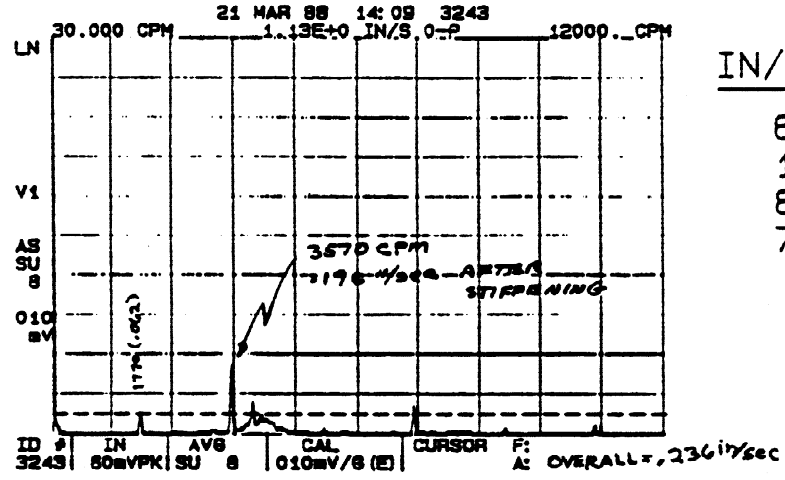
PLOT 3A.

(REPEAT OF PLOT 1)

21-MAR-88

POSITION 4V
RUNNING VIBRATION

18: 53: 29



IN/S 0-P	CPM
61.9E-3	1770.0
196.E-3	3570.0
87.7E-3	3959.9
77.4E-3	7110.0

PEAK PICK ON TRACE 1

PLOT 3B.

22-MAR-88

POSITION 4V
RUNNING VIBRATION AFTER STIFFENING

12: 30: 17

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CORRECTION OF EXCESSIVE VIBRATION TRANSMISSION INTO TWO XLE RECIPROCATING AIR COMPRESSORS DUE TO ATTACHED PIPING RESONANCE

I. BACKGROUND

Each of the two Ingersoll Rand XLE Reciprocating Compressors (Nos. 5 and 6) had been subjected to high vibration for some time which had likely contributed periodically to motor failures. Technical Associates was employed to determine the cause of high vibration, how severe it was and to make recommendations required to resolve any problems detected.

II. ANALYSIS RESULTS

Figures 1 and 2 show where vibration measurements were made on the compressors and attached piping. Figure 1 shows that the pipe passed through an exterior wall between positions 10 and 12. From there, the pipe extended to a receiver.

Vibration measurements were made both on the compressor and attached piping, each of which will be discussed below.

A. GENERAL CONDITION OF THE COMPRESSORS

Figure 3 was captured on position 3H of the #6 XLE Compressor (see Figure 2). It showed a classic spectrum for a compressor of this type with 2X RPM exceeding 1X RPM due to the fact that secondary reciprocating forces in these machines normally do cause higher vibration than the primary forces do at 1X RPM. Looking at Figure 11.08G, note that levels were .173 in/sec at 450 RPM (1X RPM) and .196 in/sec at 900 CPM (2X RPM). These levels, which had a total overall of .268 in/sec, were found quite satisfactory for a compressor of this type.

The compressors appeared to be in good mechanical condition and were properly installed with adequate inertia blocks, which are frequently found to be inadequate. Only one compressor, #5, had a slight amount of looseness between the compressor and inertia block, but was not a problem at this time. Compressor #6 had a maximum overall vibration level of .268 in/sec, taken at position 3H (Figure 2) versus a normal allowable of about .50 in/sec for compressors of this type. Compressor #5, with the looseness between the compressor and inertia block, had a somewhat higher vibration of .365 in/sec, but still quite acceptable for this machine type.

B. PIPING VIBRATION

A schematic of the exhaust piping for XLE Compressor #6 is shown by Figure 1. High vibration levels were found on the piping attached to XLE #6. Figure 4 shows such a spectrum taken at position 9 located close to the point where the pipe passed through the exterior wall (see Figure 1). Note the high levels of .568 in/sec at 450 CPM (1X RPM) and 1.55 in/sec at 900 CPM (2X compressor speed). In comparison, the harmonics above 2X RPM were very low.

Piping attached to reciprocating air compressors is often prone to resonant vibration with the forcing frequencies coming from the compressor transmitting into the piping. Due to the fact that XLE compressors normally put out noticeable vibration at both 1X and 2X RPM (due to primary and secondary reciprocating forces), it is necessary to keep natural frequencies of attached piping well away from each of these two predominant forcing

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frequencies. Not only do these compressor forcing frequencies often excite mechanical natural frequencies of the attached piping structure, pressure pulsations from the compressed air can also excite acoustical natural frequencies within the piping as well. This can be detected by placing pressure transducers within the pipe and connecting it to an FFT analyzer for spectral analysis. Therefore, a plan was made to first check for and attempt to correct any mechanical natural frequency problems encountered; then follow with a pressure pulsation check if vibration problems still remained.

In order to test for the presence of possible natural frequencies, a series of impulse natural frequency tests were conducted throughout the piping on each of the positions shown in Figure 1. Several resonances were found, dominated particularly by one at position 9 located close to the wall. Figure 5 shows the natural frequency found at position 9 which had the highest vibration throughout the piping of XLE #6. This spectrum shows that the pipe natural frequency here was 870 CPM, or only 3.3% below the compressor source frequency of 900 CPM (2X compressor speed). Similar results were found on XLE #5 compressor as seen in Figure 6. Here again, the highest vibration was at 2X RPM with a level of .743 in/sec compared to only .289 in/sec at 1X RPM (see Figure 6).

III. RECOMMENDATIONS

It was recommended that braces be installed at just those pipe locations that appeared to be resonant. Importantly, simply adding a brace to restrain a structural member such as piping will be effective if it helps eliminate resonance. However, if the structure were not resonant, adding the brace may actually increase forces and shift these forces to undesirable locations where damage may result. Therefore, it is important to ensure resonance does exist. Since it was proven to exit on the piping of both #5 and #6 compressors, recommendations were made to temporarily attach such bracing and evaluate results.

IV. RESULTS OF CORRECTIVE ACTIONS TAKEN

Figure 7 was taken on position 9 of XLE #6 after bracing was installed between the exterior wall and pipe. Note that the natural frequency was moved noticeably, decreasing from 870 CPM before bracing (Figure 5) down to 675 CPM after bracing (see Figure 7). Normally, when a brace is placed between a structure and a resonant member, it will increase the natural frequency since it will decrease operating deflection. However, in this case, the bracing not only increased stiffness, but also significantly increased mass since it effectively tied the mass of the exterior wall in with the system (natural frequency is proportional to stiffness, divided by mass or $\sqrt{k/m}$). Referring to Figure 7, note that the braced natural frequency of 675 CPM was brought right at the midpoint between the 450 CPM running speed and the 900 CPM second harmonic (a threaded rod was employed to help "tune" the brace forces in order to move the natural frequency right to this location).

Figure 8 shows the results of how much vibration was decreased on Compressor #6 by moving the natural frequency away from resonance. Referring to Figure 8, note that the vibration at 1X RPM (450 CPM) was reduced 85% from .568 in/sec down to .085 in/sec. Similarly, the vibration at 2X RPM (900 CPM) was reduced even more dramatically by 91% from 1.55 in/sec down to only .141 in/sec (Figure 8 is shown on the same sheet with Figure 4 to ease comparison, but note the great difference in full-scale values for the vertical scale).

Success was also achieved by bracing resonant sections of piping connected to XLE #5 as shown by comparing Figures 6 and 9 (each of which are shown on the same sheet for ease of comparison). Note that vibration at 1X RPM was reduced from .289 in/sec down to .175 in/sec while an even greater reduction was realized at 2X RPM decreasing from .743 in/sec down to .232 in/sec.

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The above discussion shows that the piping vibration was reduced by eliminating its resonance with the first and second harmonic running speeds of the compressor. This was achieved by changing the natural frequency of the piping to move the natural frequencies away from these forcing functions. Therefore, the bracing did prove very effective and was the correct solution.

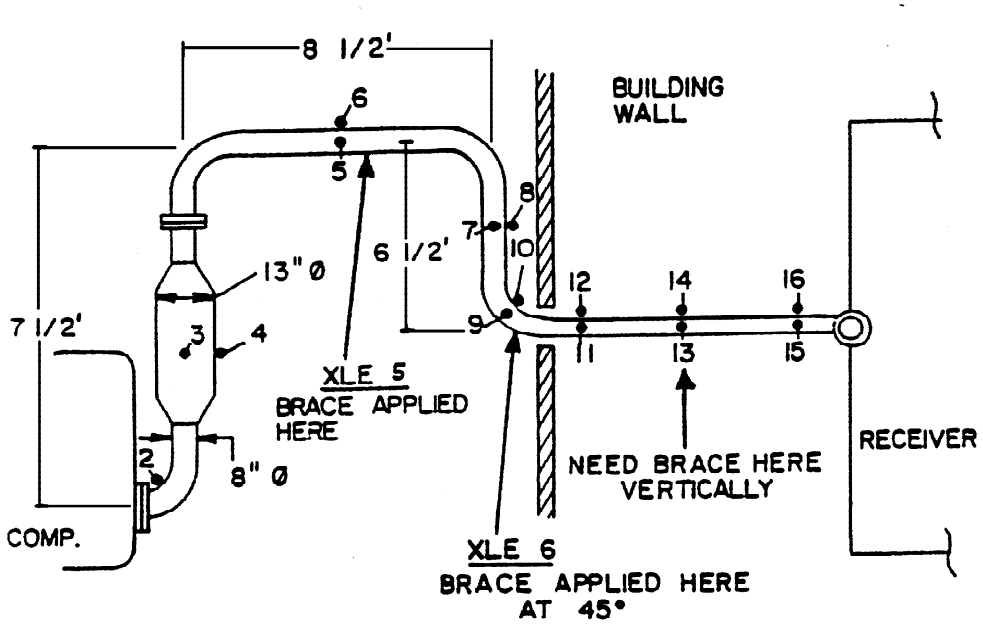

FIGURE 1
XLE COMPRESSOR #6 PIPING

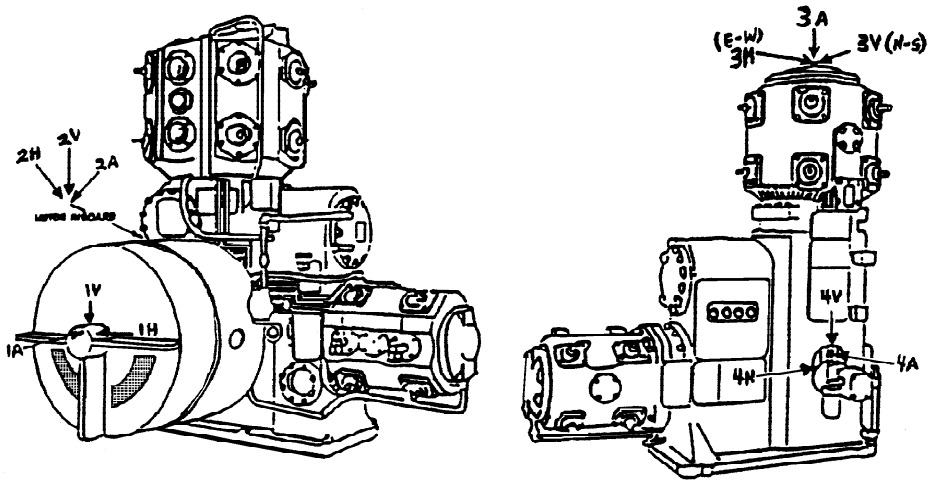

FIGURE 2
VIBRATION MEASUREMENT LOCATIONS ON INGERSOLL-RAND XLE RECIPROCATING AIR COMPRESSORS (2-STAGE)

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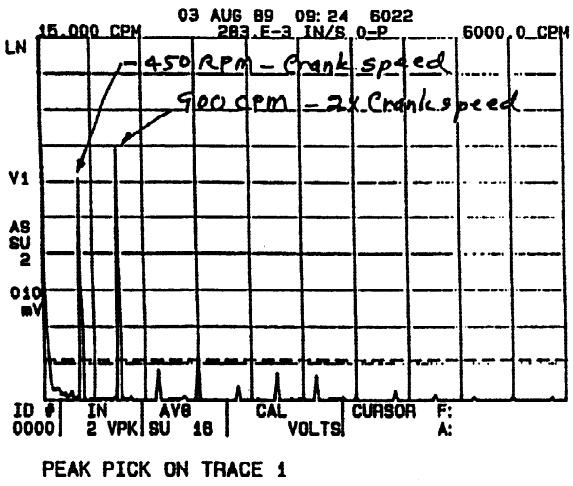


IN/S 0-P	CPM
173.E-3	450.00
196.E-3	900.00
34.9E-3	1800.0

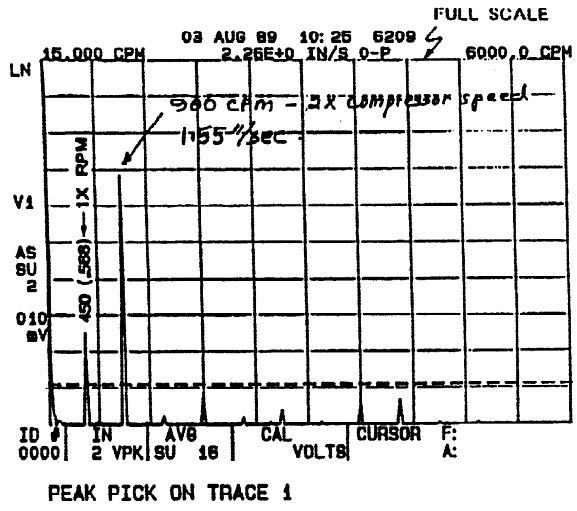
07-AUG-89 OA = .268 "/SEC

XLE COMP. # 6

08:18:28 VIBRATION AT POINT 3H (FIGURE 11.08-F)

FIGURE 3.

COMP # 6



IN/S 0-P	CPM
568.E-3	450.00
1.55	900.00

07-AUG-89 XLE COMP # 6 OVERALL = 1.67 "/sec ON EXHAUST PIPING POINT 9 - NO BRACING

09:17:07

FIGURE 4

COMP # 6

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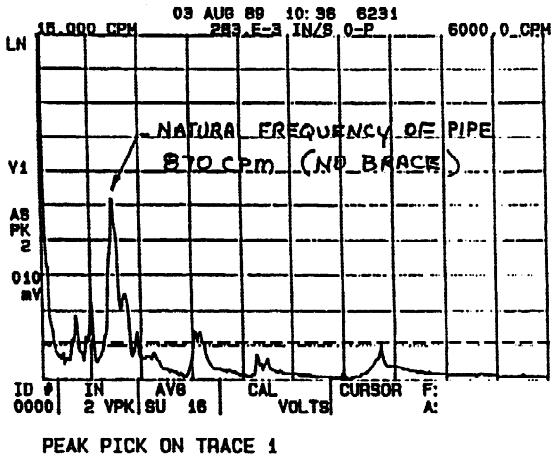


PEAK PICK ON TRACE 1

IN/S 0-P	CPM
33.6E-3	375.00
53.0E-3	420.00
34.9E-3	540.00
64.6E-3	600.00
36.2E-3	764.97
147.E-3	870.00
69.8E-3	1005.0
38.8E-3	1155.0
40.1E-3	1830.0
38.8E-3	1890.0

07-AUG-89

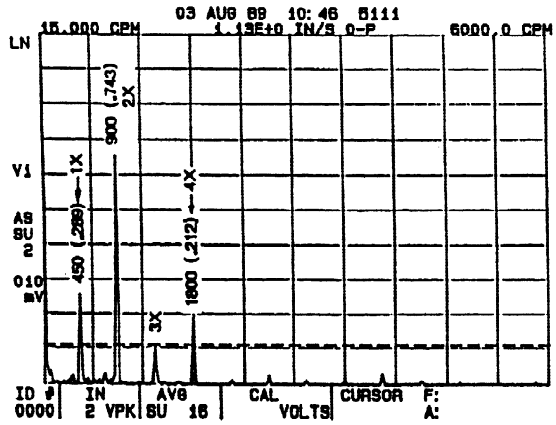
XLE COMP. # 6

09: 26: 44 IMPULSE TEST ON EXHAUST PIPING (NO BRACE)

POINT 9

FIGURE 5.

COMP # 6



PEAK PICK ON TRACE 1

IN/S 0-P	CPM
289.E-3	450.00
743.E-3	900.00
212.E-3	1800.0

07-AUG-89 OVERALL = .837 "/sec

XLE COMP # 5

10: 29: 44 PIPING VIBRATION (NO BRACE)

FIGURE 6.

COMP # 5

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CPM
165.00
450.00
675.00~
1020.0
1110.0
1365.0
1979.9
2070.0
2610.0
2715.0
4050.0
4200.0

WITH BANCE)

COMP # 6

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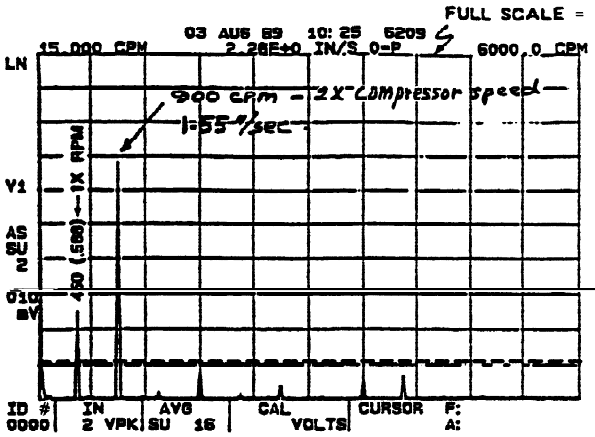


IN/S 0-P	CPM
568.E-3	450.00
1.55	900.00

PEAK PICK ON TRACE 1

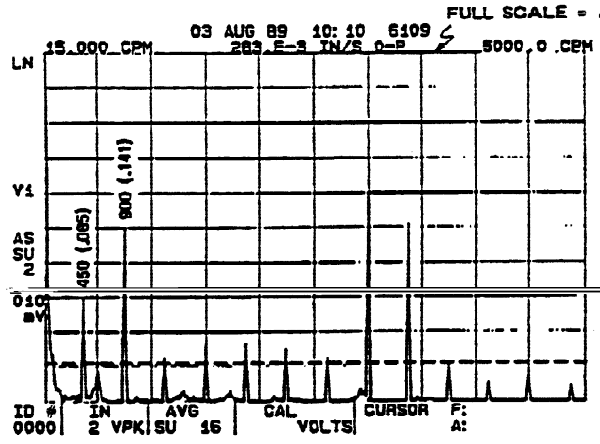
07-AUG-89

09: 17: 07

XLE COMP # 6
OVERALL = 1.67 "/sec
ON EXHAUST PIPING
POINT 9 - No BRACING

FIGURE 4.
(REPEATED FOR READER)

COMP # 6



IN/S 0-P	CPM
38.8E-3	105.00
85.3E-3	450.00
141.E-3	900.00
34.9E-3	1350.0
45.2E-3	1800.0
42.6E-3	2700.0
34.9E-3	3150.0
171.E-3	3600.0
146.E-3	4050.0

PEAK PICK ON TRACE 1

07-AUG-89

08: 59: 44

XLE COMP # 6
OVERALL = .303 "/sec
ON EXHAUST PIPING
POINT 9 - WITH BRACE

FIGURE 8.

COMP # 6

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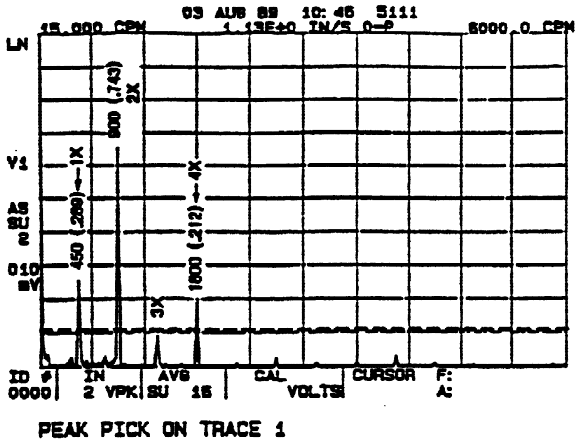


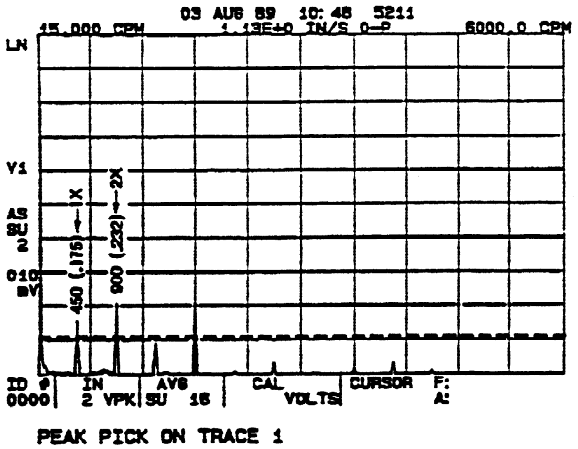
PEAK PICK ON TRACE 1

IN/S 0-P	CPM
289.E-3	450.00
743.E-3	900.00
212.E-3	1800.0

07-AUG-89 OVERALL = .837 "/sec
XLE COMP # 5
10: 29: 44 PIPING VIBRATION (NO BRACE)

FIGURE 6.
(REPEATED FOR READER)

COMP # 5



PEAK PICK ON TRACE 1

IN/S 0-P	CPM
175.E-3	450.00
232.E-3	900.00
237.E-3	1800.0

07-AUG-89 OVERALL = .399 "/sec
XLE COMP # 5
10: 30: 32 PIPING VIBRATION (WITH BRACE)

FIGURE 9.

COMP # 5

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DETECTION OF RESONANT WHIRL ON PAPER MACHINE ROLLS

I. BACKGROUND

On August 24, 1988, vibration measurements were made on the fourth dryer stretch roll to evaluate for possible in-place balancing. On August 26, the fourth dryer breaker stack lead-in roll was also evaluated for possible in-place balancing.

II. CONCLUSIONS

A. Fourth Dryer Stretch Roll

Vibration measurements on the third and fourth dryer stretch rolls were made on the support structure. The third dryer stretch roll which ran smoothly was used as a baseline to evaluate the fourth roll. Figure 1 compares the roll 3 spectrum to the roll 4 spectrum using the same full scale values for direct comparison. Roll 4 was approximately 32 times higher than roll 3! Vibration levels were undoubtedly much higher on the bearing than those values shown on Figure 1 taken on the support structure. The very high vibration of the fourth stretch roll likely caused a mechanical looseness phenomenon as shown by the multiple running speed harmonics of Figure 2 taken in the vertical direction.

It was concluded that the high vibration in roll 4 was likely caused by whirl; that is, deflection owing to operation at resonance (speed of the roll is equal to the roll natural frequency). To balance such a roll requires couples to oppose the bowing that occurs, as shown by Figure 3. The couple usually requires access to the quarter points of the roll as shown by Figure 3. In view of this, it was recommended that the roll be replaced rather than to attempt an in-place correction.

Some material was removed from the roll shell in the past and it probably has a lower natural frequency than roll 3. This would explain why roll 3, which is running at the same speed, was not whirling.

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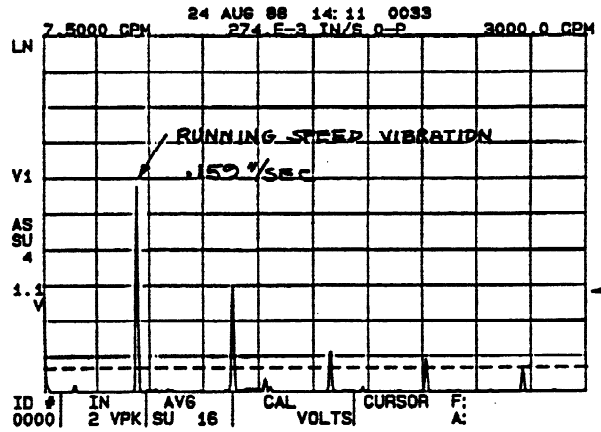
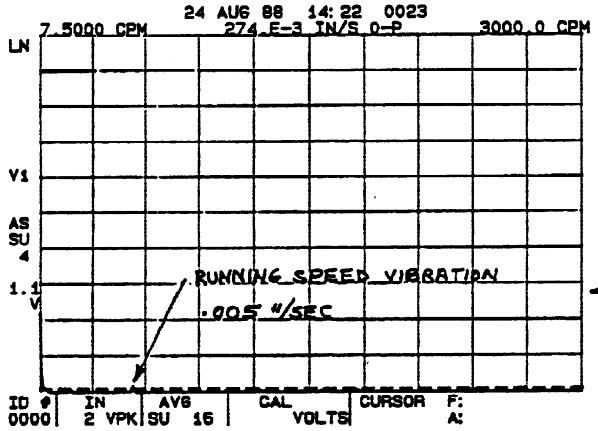


IN/S 0-P	CPM
159.E-3 1 x RPM	532.50
82.6E-3 2 x RPM	1065.0
31.3E-3 3 x RPM	1597.4
25.0E-3 4 x RPM	2122.5
18.8E-3 5 x RPM	2655.0

← FOURTH DRYER STRETCH ROLL
.274 "/SEC FULL SCALE

24-AUG-88

DRYR ROLL4
VERT.



IN/S 0-P	CPM
5.00E-3	532.50
2.50E-3	1057.5

← THIRD DRYER STRETCH ROLL
.274 "/SEC FULL SCALE

24-AUG-88

18: 29: 49

DRYR ROLL3

FIGURE 1.

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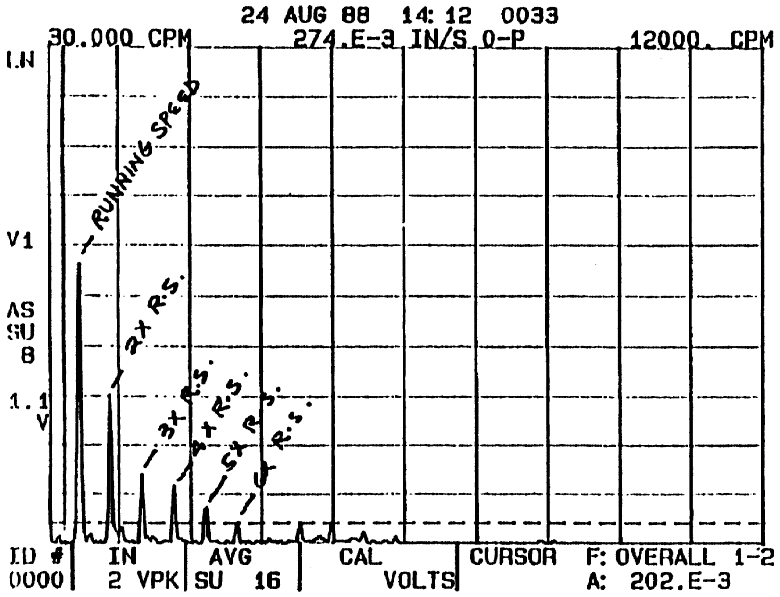


PEAK PICK ON TRACE 1

IN/S 0-P		CPM
155.E-3	1 X RPM	540.00
83.8E-3	2 X RPM	1050.0
38.8E-3	3 X RPM	1589.9
32.5E-3		2130.0
20.0E-3		2670.0
11.3E-3		3179.9
11.3E-3		4770.0

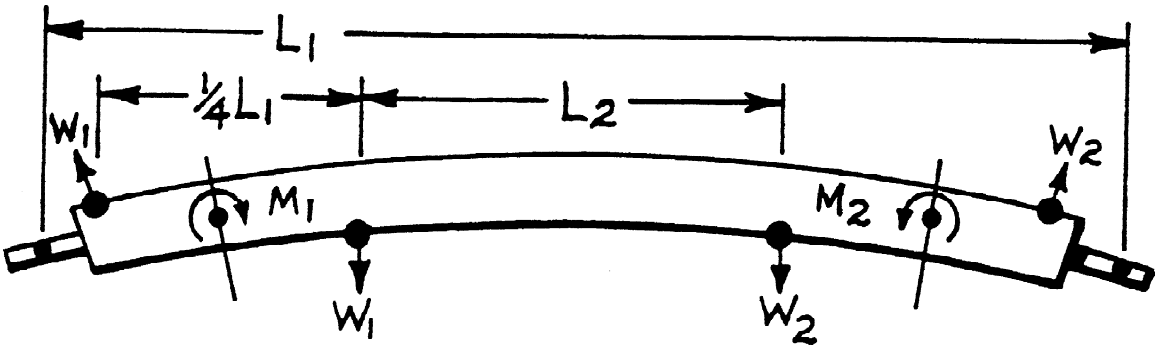
24-AUG-88

18: 21: 22

FIGURE 2.

DRYR ROLL4

FIGURE 3.



Weights W_1 and W_2 create a couple about points M_1 & M_2 to eliminate whirl. This solution is only good to eliminate whirl, where whirl is defined as a bow without stress reversals; that is, the critical speed of the roll is equal to running speed. Whip is resonance at some frequency other than running speed and requires a different solution than that shown.

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B. Breaker Stack Lead-In Roll (Dryer 4 Section)

On August 26, 1998, measurements were made on the breaker stack lead-in roll with the process down, which enabled Technical Associates to trace the roll vibration from 0 to 504 RPM and compare this to that which would be generated only by unbalance ($U = mr\omega^2$). Figure 4 shows the peak vibration trace obtained during the runup. The important feature to note on Figure 4 is the abrupt increase in vibration at approximately 500 RPM, an indication that the roll has gone into whirl. Figure 5 is an enlarged view of the same trace showing the calculated unbalance curve, the dashed line, which is based on the vibration amplitude of .147 in/sec at 165 RPM. Comparing the actual trace to the unbalance curve shows that in addition to the onset of whirl at 500 RPM, there were several points of resonant amplification throughout the operating speed range. These points of amplification were likely caused by structural resonances and would undoubtedly be greatly reduced when the roll is properly balanced. If current vibration levels continued for a long period, structural damage to the framework could occur.

The equation and tabulation below (Table I) were used to generate the unbalance curve of Figure 5.

$$U = mr\omega^2$$

$$mr = \frac{U}{\omega^2} = \frac{.147}{(.165)^2} = 5.399 \times 10^6$$

TABLE I.

RPM	$mr\omega^2$	TRACE VALUE
165	.147	.147
207	.231	.296
275	.408	.571
324	.567	.922
384	.796	1.140
417	.939	1.630
504	1.370	2.580

Note the great departure from pure roll unbalance at 504 RPM. This strongly indicated the presence of resonant amplitude amplification. Resolution of this problem would require the balance correction procedure outlined in Figure 3.

III. RECOMMENDATIONS

The fourth dryer stretch roll likely had whirl caused by roll resonance at running speed. The fourth dryer breaker stack lead-in roll likely had whirl caused by roll resonance at running speed. It was recommended that both the stretch roll and the breaker stack lead-in roll be replaced rather than attempt in-place balancing.

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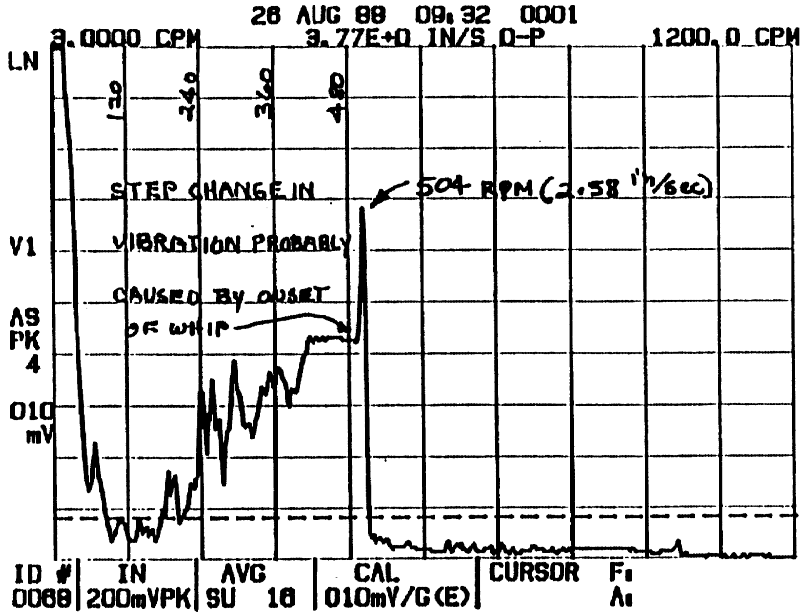
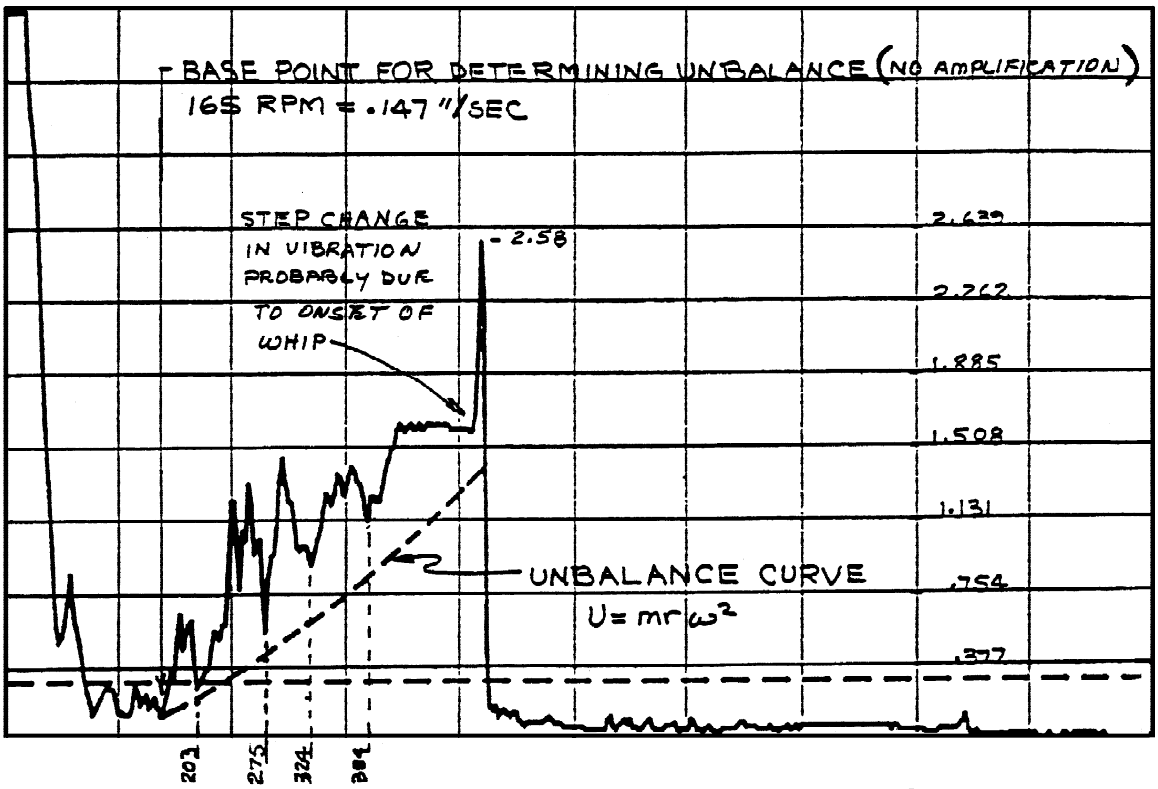


31-AUG-88

08:54:28

IN/S 0-P	CPM
861. E-3	69. 000
362. E-3	177. 00
654. E-3	185. 99
620. E-3	197. 99
568. E-3	221. 99
1. 24	240. 00
1. 08	252. 00
1. 33	258. 00
1. 03	270. 00
1. 46	294. 00
998. E-3	315. 00
1. 27	339. 00
1. 38	351. 00
1. 41	366. 00
1. 26	387. 00
1. 64	416. 98
1. 64	429. 00
1. 64	438. 00
2. 58	503. 98

FIGURE 4.



The dashed line represents an unbalance trace for this roll using .147"/SEC as vibration due to unbalance at 165 RPM. The solid line is the actual trace of vibration.

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DETECTION AND CORRECTION OF A SERIOUS ROTOR RESONANCE ON A CENTRIFUGAL COMPRESSOR WHICH HAD SUCCESSFULLY OPERATED FOR 15 YEARS BEFORE A RECENT OVERHAUL

I. BACKGROUND

The purpose of this study was to perform a real-time vibration analysis on an Ingersoll-Rand model C-40 centrifugal air compressor for identification of a possible electrical problem. The 4000 CFM compressor has a 1035 horsepower, two pole motor running at 3580 RPM and is designated as Centac #1 at this plant. The compressor was said to have had a running speed vibration amplitude of 1.2 mils on the motor bearing which would drop rapidly when shut down.

The vibration history of Centac #1 was said to have had increased over time. The motor was therefore cleaned; but then a high temperature alarm tripped out the motor, so the motor bearings were replaced. It was then observed that the motor shaft and coupling had bottomed out which prevented the motor from obtaining magnetic center, but the vibration level still remained high (previously had noted existence of a soft foot which was subsequently shimmed, but may not have been sufficient; however, the client had consecutively loosened the mounting bolts in attempts to reduce vibration levels if caused by a soft foot). Attempts were made to balance the motor in the field, but were not successful. The rotor would not respond to application of trial balance weights, no matter how much either the size or angular position was varied. Also, the phase was unstable.

Technical Associates performed real-time vibration analysis on this machine on July 27, 1990. Due to the high demand for this compressor, the results were conveyed over the phone to Plant Services the following day. As per Technical Associates' diagnosis, the motor bearings were replaced. Following this repair, the vibration levels were reported to be acceptable. This report, however, is presented as a documentation of the diagnosis methodical approach. Following is a summary of II. CONCLUSIONS and III. RECOMMENDATIONS, followed by a detailed IV. RESULTS OF CORRECTIVE ACTIONS**.**

II. CONCLUSIONS

A. MOTOR ELECTRICAL

(Please note the "Electrical Problems" section of the Diagnostic Chart page which is included in the report Appendix which describes how these electrical conditions were evaluated. This page is one of four pages of the diagnostic chart developed by Technical Associates and is used in our ongoing "Vibration Diagnostics" seminars).

1. The motor electrical condition was acceptable. Rotor bar pass frequency and slip frequency sidebands at running speed exist, indicating some signs of developing electrical problems, possibly due to a variable air gap between rotor and stator. However, much of the electrical vibration may have been mechanically induced. That is, the variable clearances in the motor bearings and the resonant operation later discussed could very likely have adversely affected the rotor orbit within the stator, thus "artificially" producing electrical vibration.

2. The rapid drop of vibration at running speed upon shutdown of power to the motor was not due to electrical problems.

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B. MOTOR ALIGNMENT

1. Slight misalignment was indicated across the coupling to the compressor. However, this problem was minor in comparison with others.

2. Misalignment was also indicated within the motor bearings and housing.

3. A soft foot may exist causing a twist in the motor housing, thereby affecting alignment of bearings and motor electrical vibration.

C. RESONANCE

Clearly, the dominant problem on the Centac #1 motor was found to be resonance of the operating speed with the horizontal natural frequency of the motor rotor/bearing support system itself. Results of the natural frequency tests are as follows:

1. System natural frequencies existed at or near 1X, 2X, and 3X motor RPM.

2. The 2X, and 3X RPM responses were reduced by welding chocks to stiffen the foundation in the horizontal direction.

3. The 1X motor RPM horizontal resonance was not affected by addition of braces which indicated this resonance to be a coupled system response across the rotor/ bearing/housing interface itself as opposed to the motor frame/foundation. This was surprising since it is unusual for a rotor natural frequency to be so near the operating speed. This was particularly perplexing when it was learned that this same motor had operated successfully for over 10 years which would be impossible if it had such a rotor natural frequency so near resonance with operating speed. However, after testing was completed, it was learned that only recently had the high vibration started after the motor bearings were replaced. Since it appeared so definite that a rotor resonance itself was involved, it was concluded that there must have been a problem with the bearing replacement which resulted in a change in bearing support stiffness. This would directly alter the natural frequency. Therefore, Technical Associates phoned the plant with the recommendations shown in Section III over the weekend so that needed corrective actions could be taken by the plant by the first of the week. Results of these actions are summarized in Section III and are elaborated on in Section IV RESULTS OF CORRECTIVE ACTIONS.

III. RECOMMENDATIONS

Technical Associates made the following recommendations to the plant over the weekend of July 28-29 so that corrective actions could commence on Monday, July 30:

A. Check motor bearings then in place for proper clearances, material selection, alteration in the design from the original alignment, and motor housing rigidity. Also check the shaft journal areas to ensure they were still in good condition and within specifications.

B. Replace the motor bearings and ensure that they are carefully blued and scraped to provide good contact between the bearing and journal (minimum 80% to 90% contact). Then carefully check clearances to ensure they were within specifications.

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IV. RESULTS OF CORRECTIVE ACTIONS

After the plant followed the above recommendations, it is understood that problems were confirmed with the inboard motor bearing clearances (.004" one end and .009" on the other end of the same bearing). This explained why it affected the motor bearing support stiffness and, therefore, rotor natural frequency. Follow-up measurements were requested by and faxed to Technical Associates for review using the client's vibration instruments. They showed a significant drop of over 86% in operating speed vibration on this motor, from an excessive .315 in/sec before to a much better level of .045 in/sec which successfully resolved these bearing and natural frequency problems (see Plot 4 taken before and Plot 15A taken after corrective actions).

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TABLE II. VIBRATION AMPLITUDE ON MOTOR AFTER BEARING REPLACEMENT OF 8-2-90 (in/sec Peak)

POSITION	3600 CPM			7200 CPM			OVERALL		
	7-27-89	7-27-90	8-2-90	7-27-89	7-27-90	8-2-90	7-27-89	7-27-90	8-2-90
1A	.047	.017	.006	.025	.059	.018	.059	.065	.033
1H	.114	.286	.062	.052	.028	.054	.133	.289	.074
1V	.060	.063	.006	.055	.020	.026	.087	.074	.038
2A	.023	.033	.011	.041	.033	.019	.060	.103	.038
2H	.258	.337	.046	.053	.067	.082	.280	.347	.089
2V	.078	.105	.011	.033	.091	.016	.096	.163	.035

7-27-89 Survey, Motor Coupled/TA

7-27-90 Diagnostic, Motor Solo/TA

8-2-90 Bearing Repl., Motor Solo/

TABLE III. PHASE READINGS FOR CENTAC #1 (104 AMPS LOADED)

POSITION	PHASE ANGLE (DEGREES)	PHASE DIFF. (DEGREES)	VIBRATION (ips)
Axial 1A	119		.071
2A	162	43	.097
3A	- 71	127	.046
Horizontal 1H	- 143		.150
2H	- 1	142	.492
3H	- 165	164	.021
Vertical 1V	- 82		.009
2V	- 41	41	.099
3V	85	126	.084

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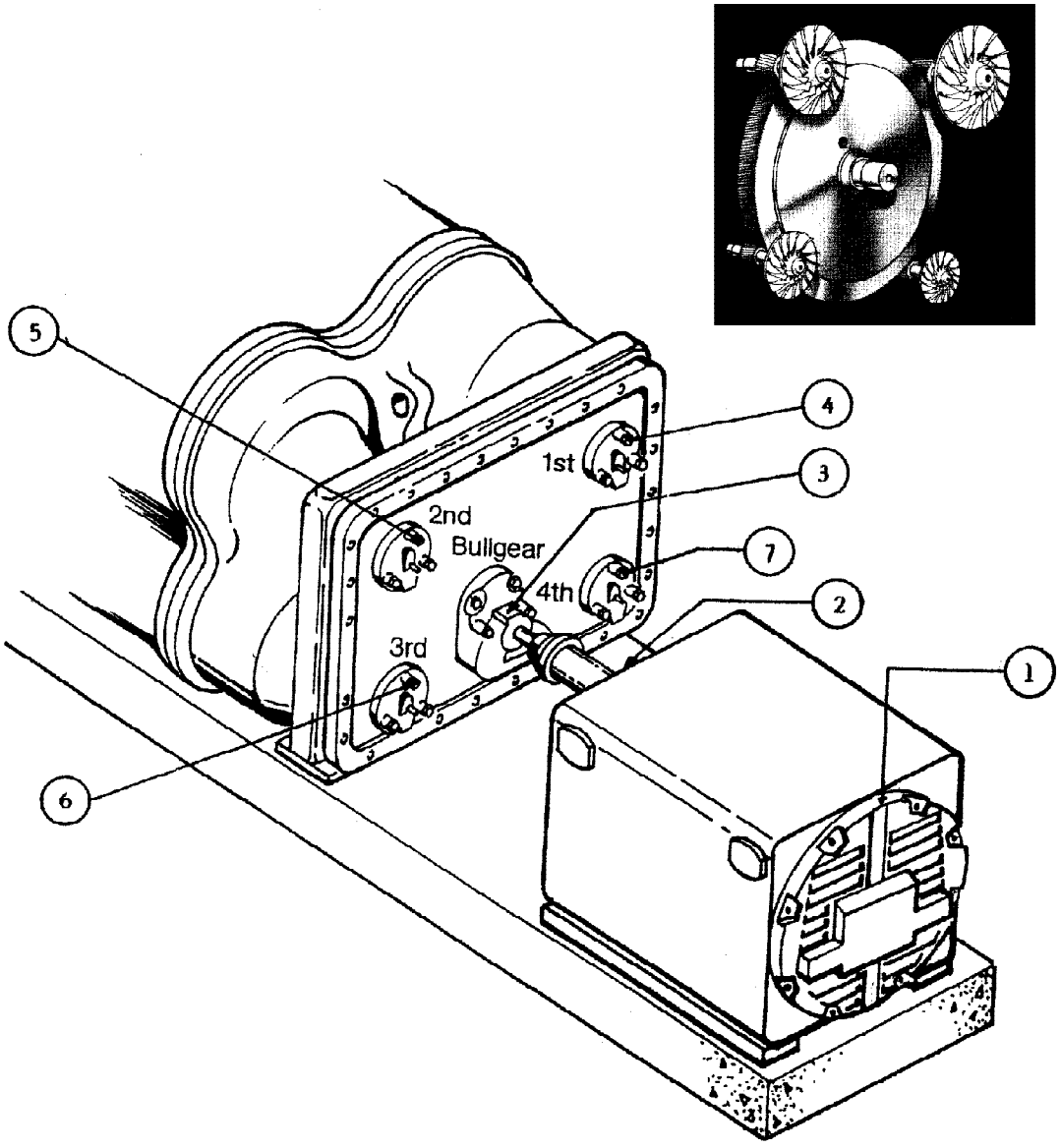

FIGURE 1: ACCELEROMETER LOCATIONS ON AN INGERSOLL-RAND CENTAC CENTRIFUGAL AIR COMPRESSOR (MODEL C-21 2100 ICFM)

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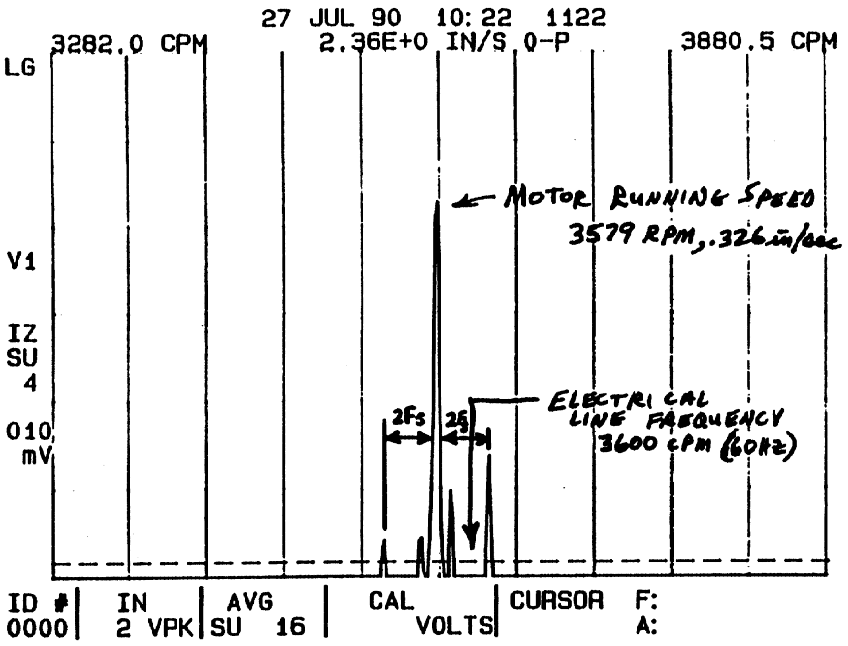

PEAK PICK ON TRACE 1

31-JUL-90

17: 11: 59

IN/S 0-P	CPM (±1.5)	
3.54E-3	3538.5	
3.66E-3	3567.0	40.5 CPM = 2 × SLIP
326.E-3 ← 1× MOTOR	3579.0	
6.78E-3	3589.5	40.5 CPM = 2 × SLIP
10.7E-3	3619.5	

MOTOR RUNNING SPEED IS SIDEBANDED BY TWICE THE SLIP FREQUENCY

F_s = NOMINAL RPM (3600) − ACTUAL RPM (3579.0) = 20 cpm

$2F_s$ = 40 cpm

POSITION 2-H

PLOT 1

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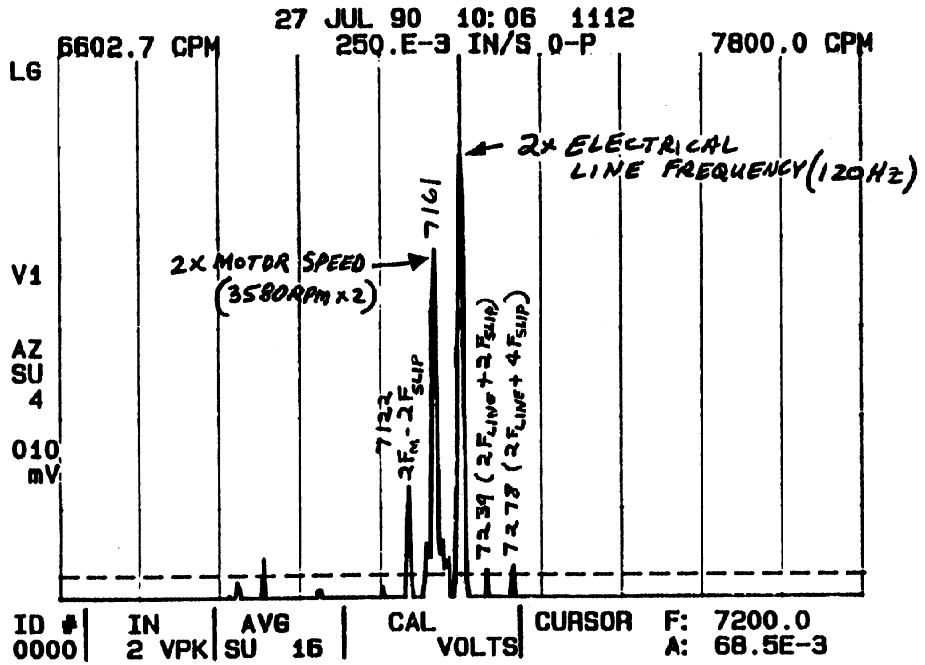

PEAK PICK ON TRACE 1

IN/S 0-P	CPM
387.E-6	6905.7
963.E-6	7122.0
471.E-6	7148.7
21.2E-3 -2X MOTOR	7161.0
486.E-6	7173.0
387.E-6	7182.0
70.6E-3 -2X LINE	7200.0
329.E-6	7239.0
351.E-6	7278.0

31-JUL-90

17:07:38

POSITION 1-H

PLOT 2

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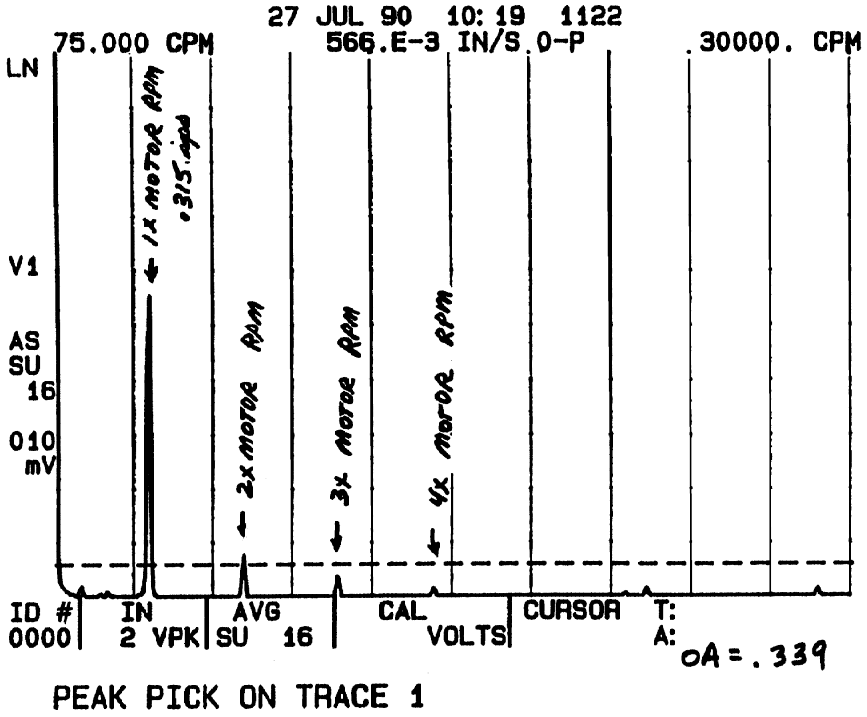

PEAK PICK ON TRACE 1

IN/S 0-P	CPM
315.E-3	3600.0
43.9E-3	7200.0

28-JUL-90

15: 27: 13

POSITION 2-H

INBOARD MOTOR BEARING, HORIZONTAL

PLOT 4

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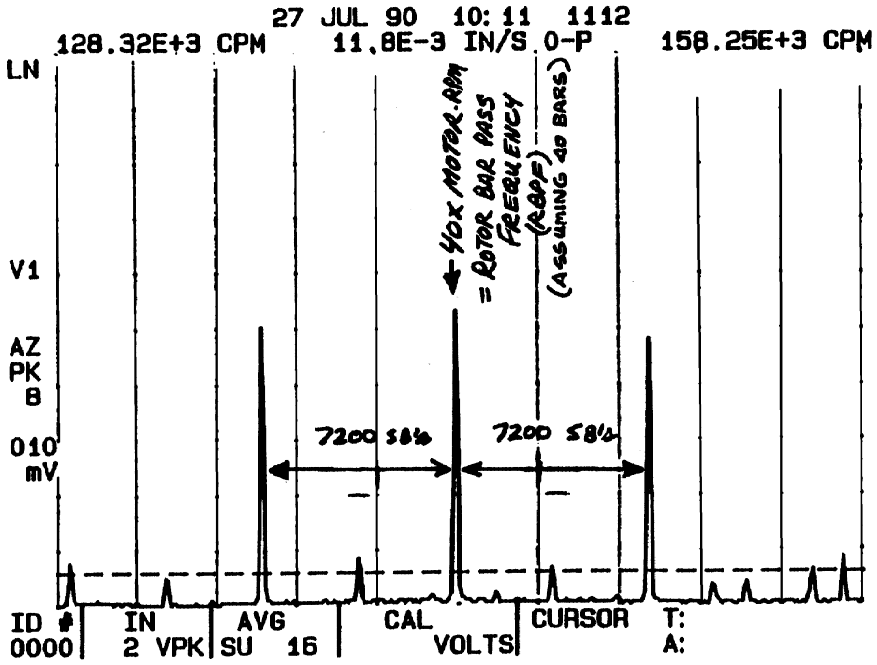

PEAK PICK ON TRACE 1

IN/S 0-P		CPM
916.E-6		128.77E+3
5.98E-3		135.97E+3
1.02E-3		139.57E+3
6.30E-3	40X MOTOR RPM	143.17E+3
808.E-6		146.78E+3
5.71E-3		150.37E+3
754.E-6		156.45E+3
1.02E-3		157.58E+3

28-JUL-90

15: 17: 52

PLOT 5

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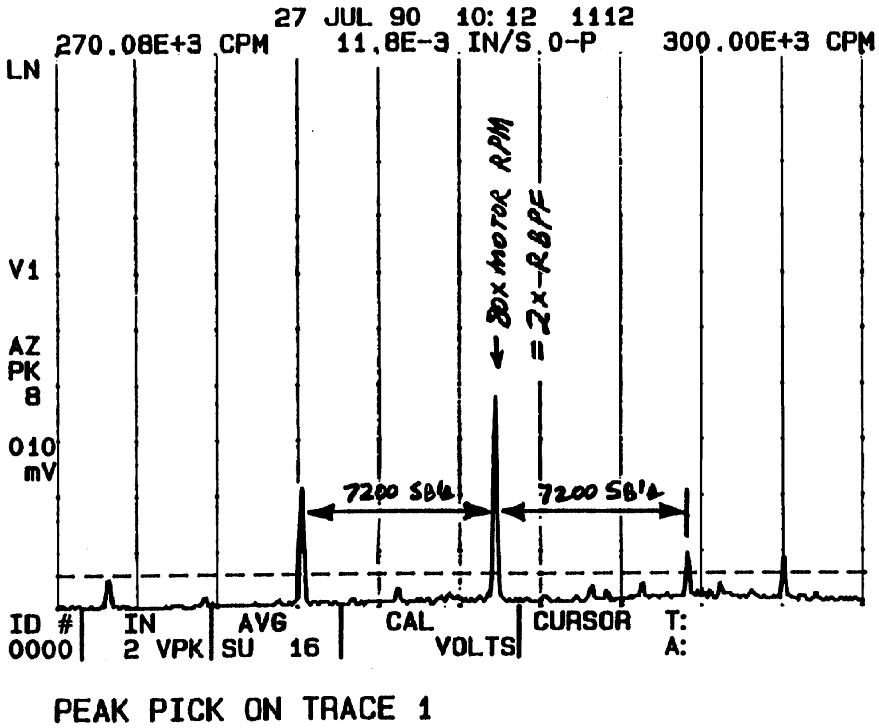

PEAK PICK ON TRACE 1

IN/S 0-P	CPM
2.53E-3 -	279.15E+3
4.47E-3 - 80x MOTOR RPM	286.35E+3
1.13E-3 -	293.47E+3
1.02E-3	297.07E+3

28-JUL-90

15: 19: 20

PLOT 6

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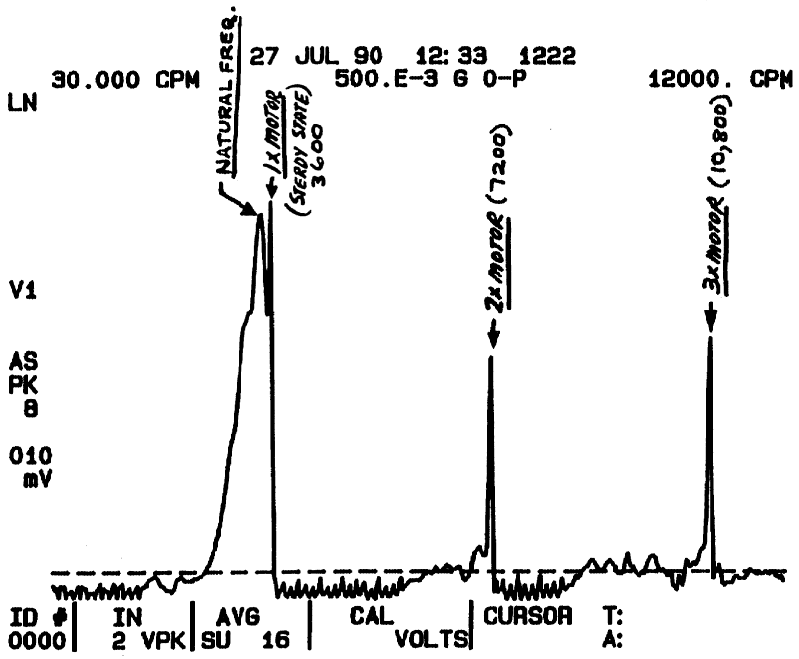

PEAK PICK ON TRACE 1

28-JUL-90

16: 13: 55

COAST-DOWN

A- POSITION 2-H
MOTOR-COUPLED - NO AMP LOAD
(PEAK HOLD SPECTRUM)

G 0-P		CPM(±30cpm)
379.E-3	- RESONANCE	3420.0
390.E-3	- 1X MOTOR (STEADY STATE)	3600.0
29.7E-3		6180.0
34.2E-3		6300.0
36.5E-3		6480.0
34.2E-3		6660.0
54.8E-3		7019.7
240.E-3	- 2X MOTOR (STEADY STATE)	7200.0
41.1E-3		8850.0
41.1E-3		9150.0
47.9E-3		9450.0
45.7E-3		9840.0
32.0E-3		10020.
41.1E-3		10410.
38.8E-3		10530.
258.E-3	- 3X MOTOR (STEADY STATE)	10799.
36.5E-3		10950.
32.0E-3		11460.
29.7E-3		11700.
29.7E-3		11820.

PLOT 7

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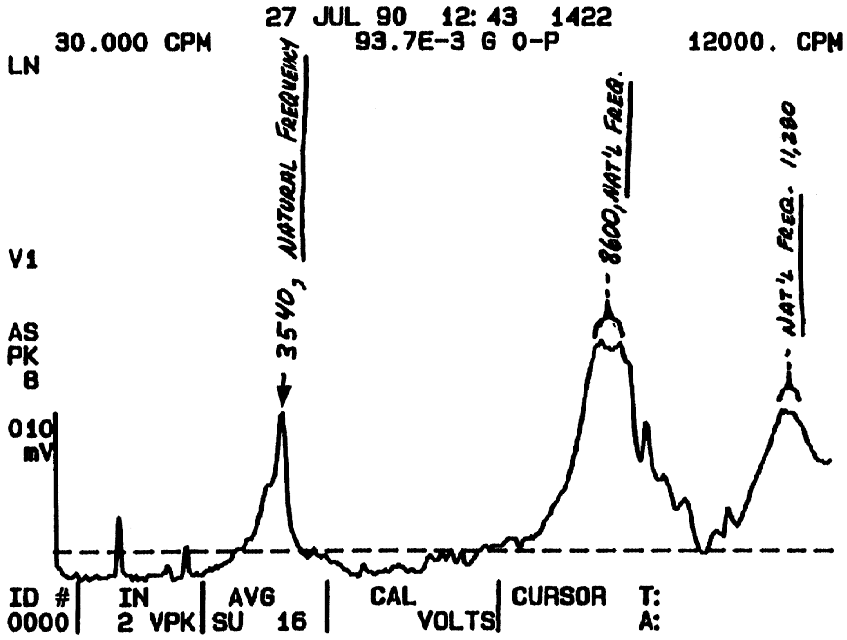

PEAK PICK ON TRACE 1

28-JUL-90

16: 15: 31

Bump

MOTOR COUPLED

F- COUPLING IMPACT, HORIZ.

A- POSITION 2-H RESPONSE

HORIZONTAL NATURAL FREQUENCIES

G 0-P		CPM (±30)
11.6E-3	- BACKGROUND EQUIPMENT	1020.0
6.42E-3	- BACKGROUND EQUIPMENT	2070.0
29.9E-3	- NAT'L FREQ.	3540.0
5.99E-3		4019.9
5.56E-3		6149.7
6.42E-3		6660.0
8.13E-3		7379.7
42.8E-3	- NAT'L FREQ.	8460.0
42.4E-3		8760.0
28.2E-3		9150.0
18.8E-3		9420.0
14.5E-3		9750.0
8.99E-3		10230.
12.8E-3		10410.
30.4E-3	- NAT'L FREQ.	11249.
30.4E-3		11310.

PLOT 8

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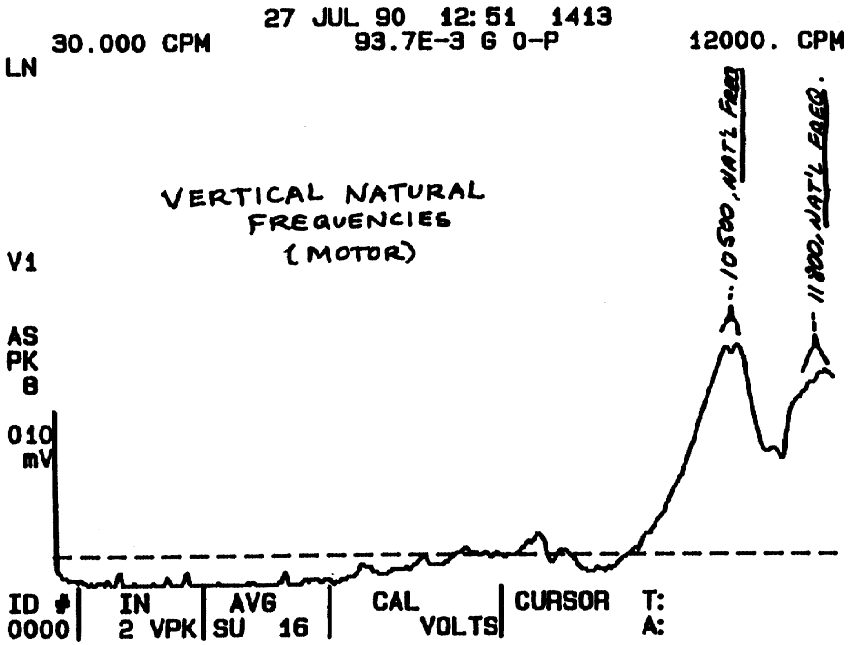

PEAK PICK ON TRACE 1

G 0-P	CPM
6.85E-3	6329.7
5.99E-3	6629.7
5.99E-3	6810.0
8.13E-3	7319.7
9.41E-3	7439.7
6.85E-3	8940.0
42.4E-3	10500.
38.1E-3	11879.

28-JUL-90

16: 21: 39

Bump

MOTOR COUPLED

F- COUPLING IMPACT, VERTICAL

A- POSITION I-V RESPONSE

PLOT 10

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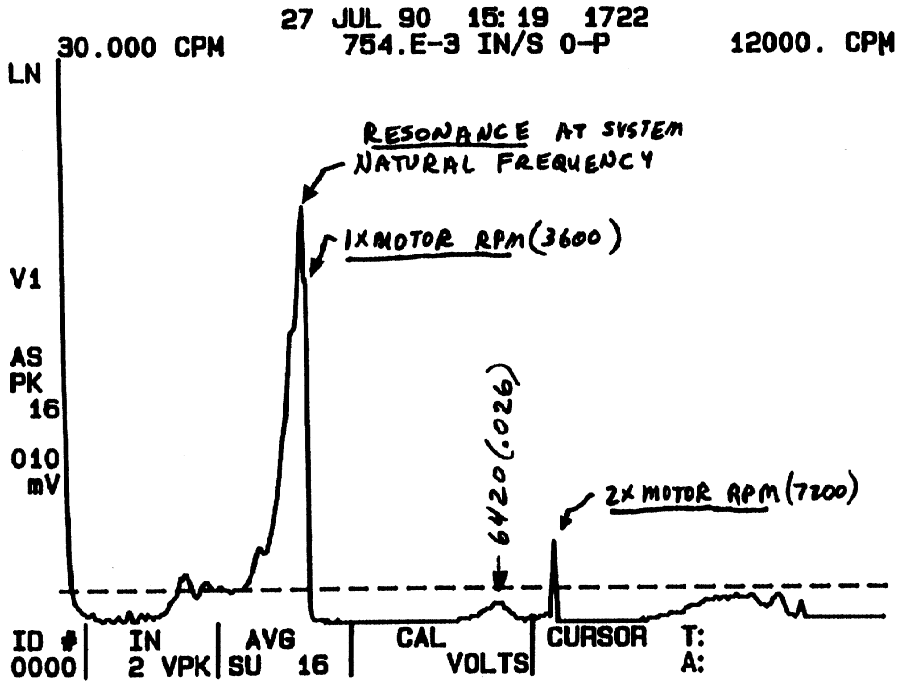

PEAK PICK ON TRACE 1

IN/S 0-P		CPM
62.0E-3		1770.0
65.4E-3		1830.0
55.1E-3		2130.0
48.2E-3		2340.0
99.8E-3		2910.0
554.E-3	- RESONANCE	3540.0
458.E-3	- 1X MOTOR (STEADY STATE)	3600.0
107.E-3	- 2X MOTOR (STEADY STATE)	7200.0

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16: 44: 55

COAST-DOWN

MOTOR SOLO, BRACED w/ WOOD + JACK

A- POSITION 2-H RESPONSE

PLOT 13

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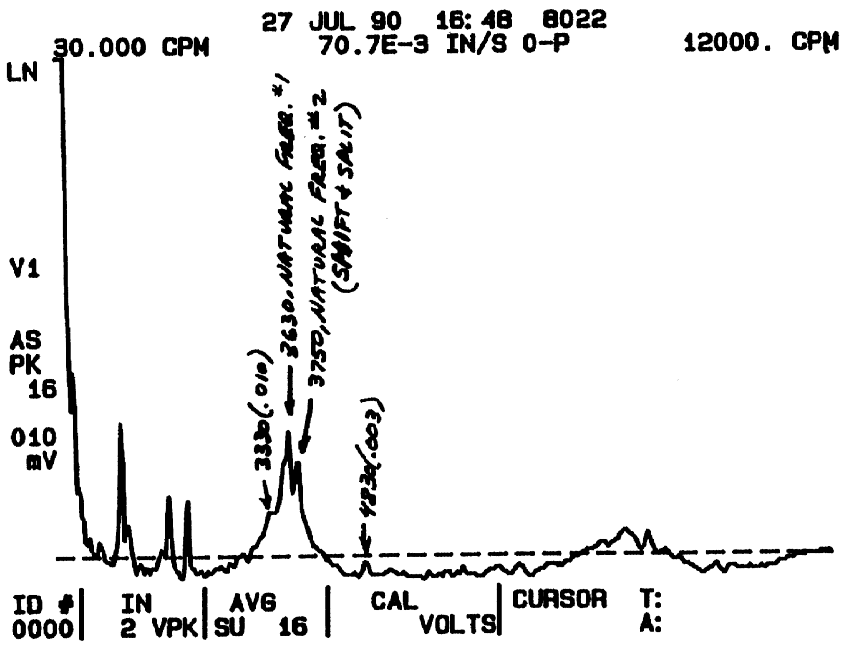


PEAK PICK ON TRACE 1

IN/S 0-P		CPM
29.1E-3		270.00
13.6E-3		360.00
6.78E-3		540.00
6.13E-3		690.00
22.3E-3	- BACKGROUND EQUIPMENT	1020.0
8.39E-3	-	1110.0
5.17E-3		1649.9
12.3E-3	- BACKGROUND EQUIPMENT	1770.0
11.6E-3	- BACKGROUND EQUIPMENT	2070.0
4.52E-3		2910.0
21.0E-3	- NAT'L FREQ. #1	3630.0
16.8E-3	- NAT'L FREQ #2	3750.0
7.43E-3		8820.0
7.10E-3		9180.0
4.84E-3		9450.0

28-JUL-90

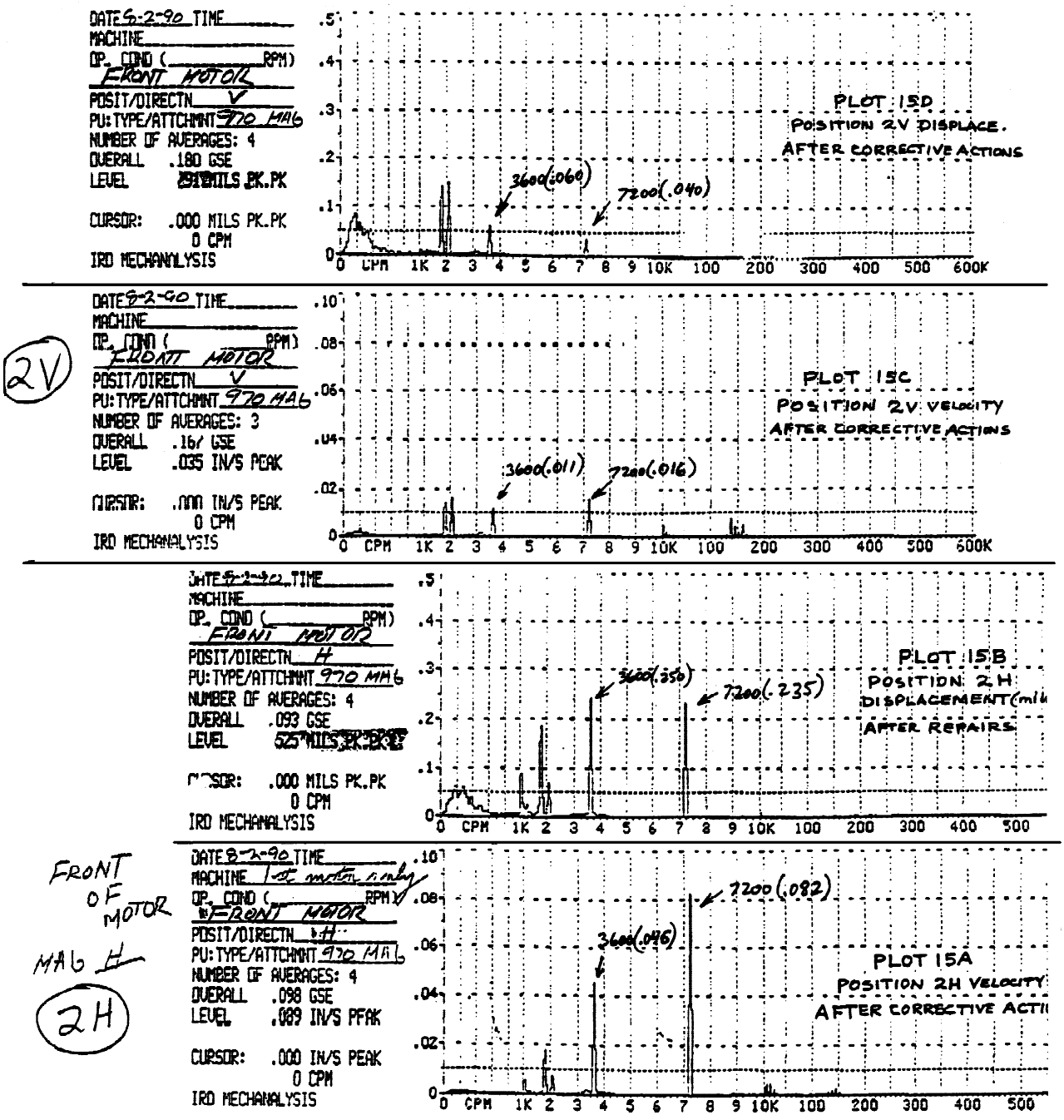
16: 49: 14

Bump

MOTOR SOLO,
WELDED BRACES

F - MOTOR HOUSING, HORIZONTAL
A - POSITION 2-H RESPONSE

PLOT 14



PLOT 15

AFTER CORRECTIVE ACTIONS

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HOW RESULTS OF COASTDOWN NATURAL FREQUENCY TESTS WERE ABLE TO GREATLY ENHANCE THE QUALITY AND OPEN UP A NEW MARKET FOR A MACHINE WITH SERIOUS RESONANCE PROBLEMS

I. BACKGROUND

Consultation services were provided by Technical Associates to a synthetic fibers client who encountered serious vibration problems with the Godet Roll Assembly pictured in Figure 1. Rolls 3A and 3B (each of which were heated) were supposed to be capable of operating up to 6000 RPM at a production rate of approximately 3000 meters/min (MPM). Unfortunately, when they tried to operate the panel at speeds just above 4000 RPM (2000 MPM), they continued to have high vibration problems which resulted in product quality problems. The client desired Technical Associates to (1) determine the cause of such high vibration at operating speeds far below that rated for the machine; and (2) submit recommendations which would resolve this problem and truly allow them to operate the machine up to 6000 RPM.

II. RESULTS AND CONCLUSIONS

Figure 2 shows results of the coastdown test performed on roll 3A of Figure 1. Note the presence of several apparent natural frequencies predominantly at about 2730, 4250 and 6525 RPM. Also note the calculation of the amplification factor (Q) and damping factor for the 6525 RPM natural frequency. Here, Q was found to be 17.9 and the damping ratio approximately .056. Note how each of these was calculated in Figure 2. A Q factor of 17.9 means that the vibration at this resonance was almost 18 times higher than that which would be present if there were no resonant amplification at this operating speed.

Next, please refer to Figure 3 which shows results of a controlled runup test for Godet Roll 3A. Several interesting things appear on the waterfall of Figure 3. First, one can see the resonances at about 2730 CPM, 4250 CPM, 5978 CPM and then in an entire region from approximately 6400 to 7100 CPM. This plot also shows something else very unusual that began to occur at a speed of approximately 6300 RPM. Note the subsynchronous frequencies that began to appear at this point and remained to a speed of approximately 6800 RPM. While one set of subsynchronous frequencies was increasing in frequency with an increase in speed, another subsynchronous set was actually decreasing with increasing rotor speed. Later, this was determined to be an instability of the Godet Roll bearings. After much study, it was determined that the bearings had improper preload (exerted by wavy springs) and also improper interference fit. Therefore, when the rotor began going into the resonant amplification zone between 6400 and 7100 RPM, this bearing instability would occur.

Figure 4 brought home the point to the client how serious a vibration problem he had. First, a test was conducted to determine the amount of residual unbalance remaining within the Godet Roll. Then, the amount of vibration which the rotor should have had (without resonant amplification) could be calculated at any speed by knowing the non-resonant vibration at several points along with knowing that unbalance vibration force ($F_{UB} = mr\omega^2$). This unbalance curve was then plotted on Figure 4 versus the actual overall vibration response curve shown in the figure. This plot then showed a great disparity between the vibration response the roll should have had were it non-resonant and what it actually experienced in the region extending from 2000 up to 7500 RPM. In addition, after review of several natural

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frequency tests on the system, it was determined that the resonances at approximately 2730 and 4250 CPM were probable frame natural frequencies, while the resonant amplification zone between approximately 6400 and 7100 CPM was due to rotor resonance. It was also confirmed that there was significant transmission of vibration from adjacent rolls into roll 3A which made it mandatory that isolation be placed within the framework between this roll and surrounding rotors. Returning to Figure 4, please notice the annotation comparing actual vibration versus theoretical unbalance vibration at 7000 RPM. At this speed, the actual overall was .320 in/sec compared to a theoretical of only .035 in/sec. The difference between .320 and .035 in/sec was attributable to resonant amplification (a factor of almost 10X).

Because of the serious rotor amplification in the area of 6400 to 7100 CPM, it was decided to build a finite element computer model of Godet Roll 3A using results learned about the structure from the natural frequency tests previously conducted. A representation of the finite element model is shown in Figure 5. Looking at this figure, note that it shows a one fourth section view of the roll (that is, you are looking at 1/4 of the roll in this picture). This was done to conserve the number of required elements within the computer model since the roll was axisymetric.

After constructing the finite element model, a number of computer runs were performed. Figure 6 shows the first 4 mode shapes of the original shaft and roll design. It predicted these modes at 98 Hz, 263 Hz, 536 Hz and 1109 Hz, respectively. It also shows the deflected shape the rotor would take at each of the modes. Using the model, a number of different iterations were performed, "artificially" changing dimensions and even materials of a number of components. Finally, Figure 7 shows that one of the simplest solutions was to construct the roll in roughly the same dimensions, but to build it of aluminum rather than steel material. Doing so would raise the fundamental natural frequency from 98 Hz (5880 RPM) up to 142 Hz (8520 RPM) (Figure 7). Such an aluminum prototype was built and the results will follow.

III. RECOMMENDATIONS

A number of recommendations were made concerning this entire Godet Roll Panel Assembly including each of the following:

A. It was recommended that specifications be completed for an isolation system for the roll/ motor assembly and surrounding frame to prevent the significant transmission of vibration from one roll to the other. It was hoped that this would not only minimize the transmission of vibration from one roll to the other, but also would eliminate excitation of many of the structural natural frequencies pictured in Figure 4.

B. It was recommended that an aluminum prototype roll be built and evaluated to confirm that it might move the rotor natural frequencies well above the desired maximum operating speed of 6000 RPM. This was based on results of the finite element computer model.

C. After a series of trials, recommendations were likewise made for the proper interference fit of Godet Roll bearings as well as their axial preloading.

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IV. RESULTS OF CORRECTIVE ACTIONS

Figure 8 shows the waterfall spectrum up to 10,800 RPM after modifications were made to the Godet Roll. Note the complete elimination of the structural resonance problems and also note that the roll fundamental natural frequency was successfully moved up from about 6000 up to 8520 RPM (from 100 to 142 Hz). In addition, even though the rotor had the 8520 RPM natural frequency, it had much lower amplitudes of only .260 in/sec even when resonant due to superior dynamic balancing by the vendor.

Interestingly, note that none of the bearing instability previously encountered at speeds of only 6100 RPM occurred until a speed of about 10,800 RPM was attained. Strangely, at this point, a subsynchronous frequency at exactly 2X the bearing cage defect frequency would occur almost every time this was attempted (cage frequency was approximately .411 times RPM). At this time, the predominant vibration would not be at 1X rotor RPM, but at the 2X bearing cage frequency. Reasons for instability at this particular frequency were never completely understood. However, as long as roll speeds remained below 10,000 RPM, this bearing instability would not occur.

Addition of the isolation material within the frame was highly effective in preventing vibration transmission from one roll to another as well as virtually eliminating the excessive vibration that was approaching 1.0 in/sec from passing into the roll when its support frame experienced resonance (at 2730 and 4250 CPM in Figure 3). Comparing Figure 3 ("Before") with Figure 8 ("After"), note that the Godet Roll now ran very smoothly all the way up until it approached its rotor resonance at 8520 CPM, with its response approaching the "Unbalance Amplitude Response Curve" of Figure 4.

The most important success with regard to the client was that they now had a machine which could successfully run not only up to the desired minimum of 6000 RPM, but now could even run up to as high as 8000 RPM with amplitudes remaining just below .200 in/sec. In addition, the vendor now had a machine which could enter a whole new market up to speeds of 8000 RPM which corresponded to production speeds of about 4200 MPM.

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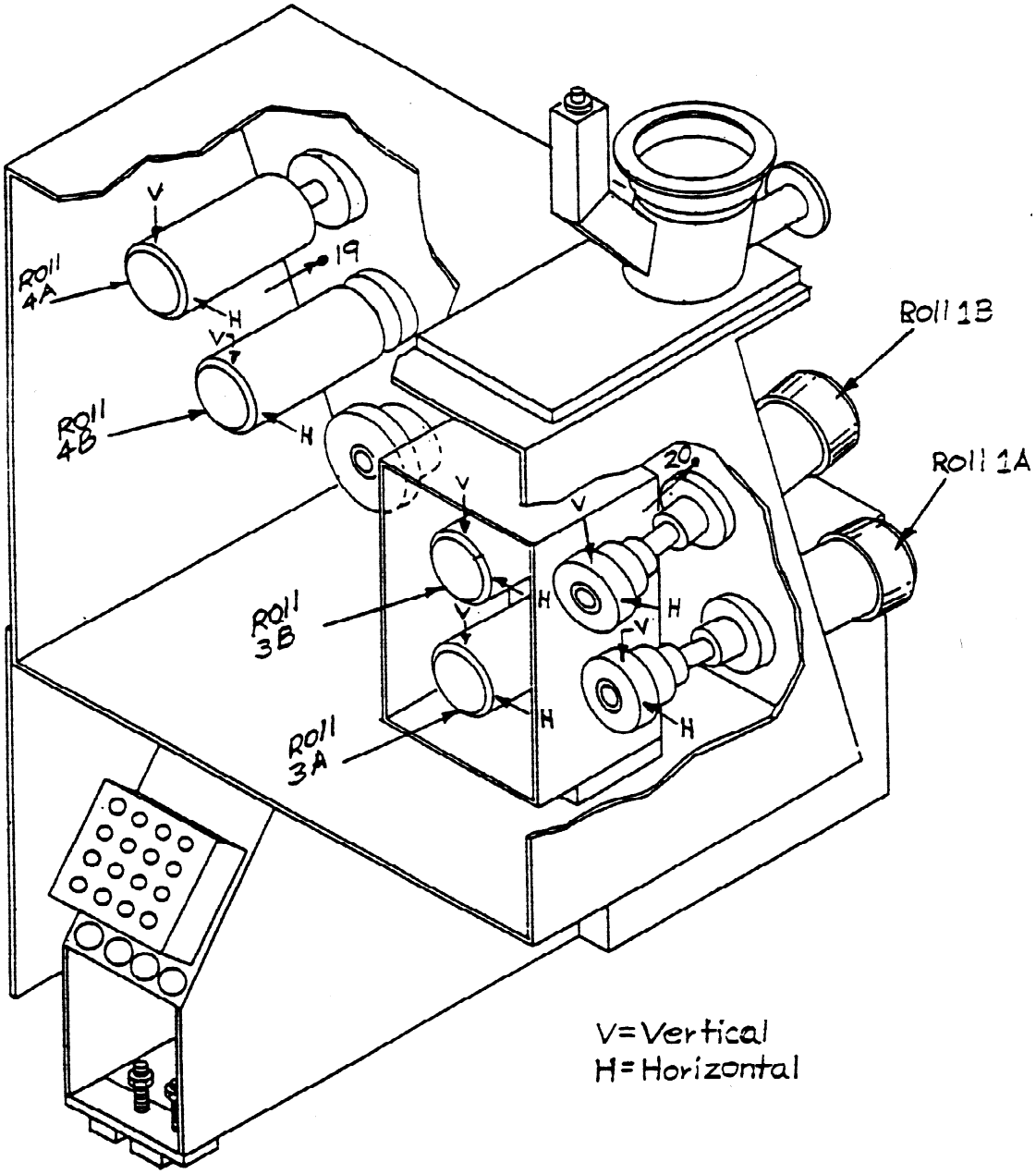

FIGURE 1.

GODET ROLL PANEL NATURAL FREQUENCY MEASUREMENT POSITIONS

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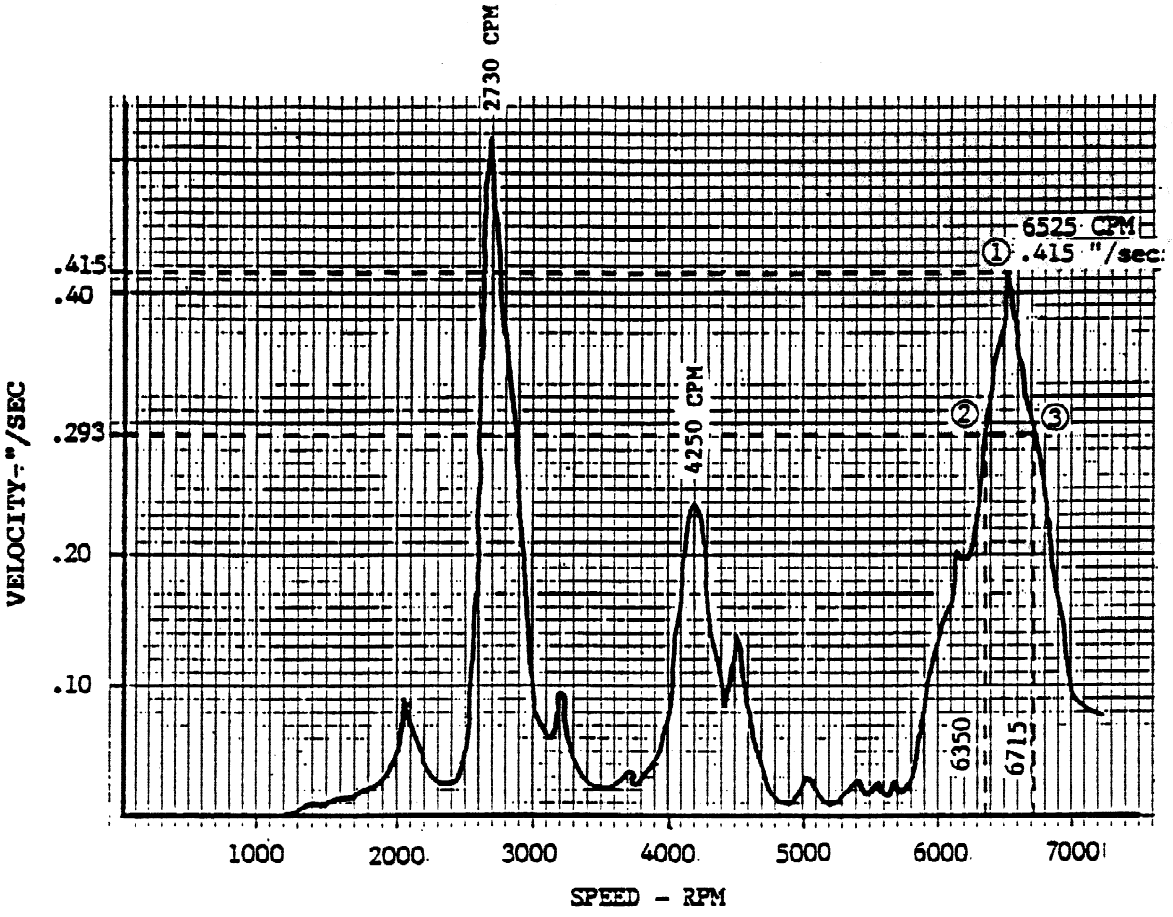
DAMPING RATIO FROM COASTDOWN VIBRATION



1 - At peak amplitude of .415 "/sec the frequency is 6525 CPM

2 - The half power points are .415 x .707 = .293 "/sec.

3 - The frequencies at the half power points are 6350 CPM and 6715 CPM

4 - The amplification

$$Q = \frac{6525}{6715 - 6350} = 17.9$$

5 - The damping ratio:

$$Q = \frac{1}{\sqrt{\left[1-\left(\frac{\omega}{\omega n}\right)^2\right]^2+\left(\frac{c}{Cc}\frac{\omega}{\omega n}\right)^2}} \qquad \text{at resonance} \quad Q = \frac{1}{2\frac{c}{Cc}\frac{\omega}{\omega n}}$$

$$\frac{C}{Cc} = \frac{1}{2Q} = \frac{1}{2(17.9)} = .056$$

FIGURE 2.

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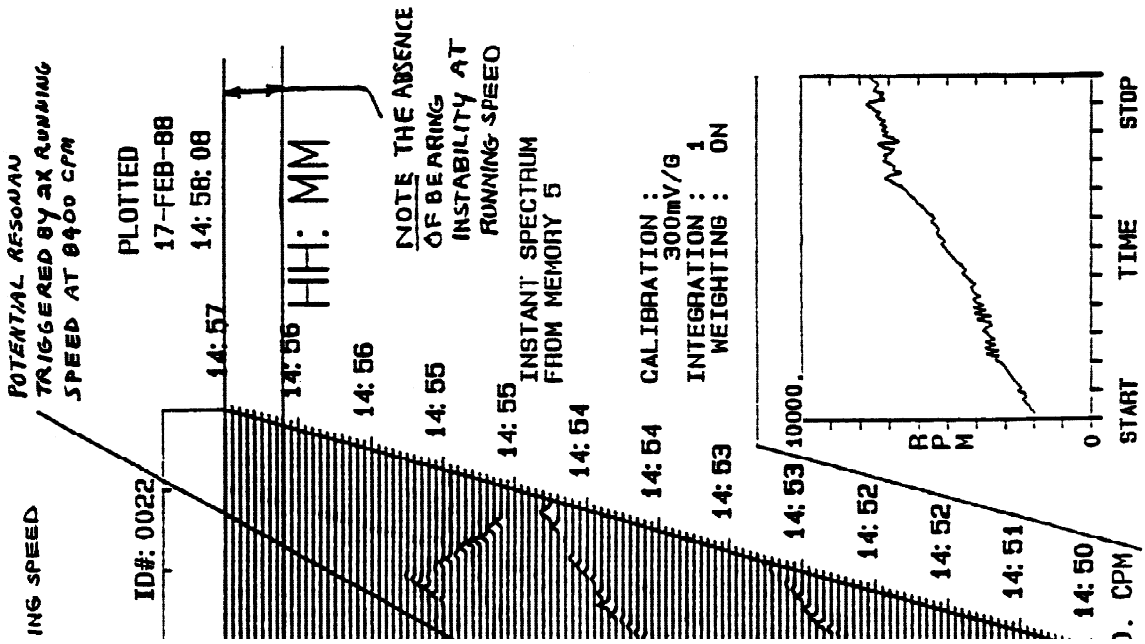

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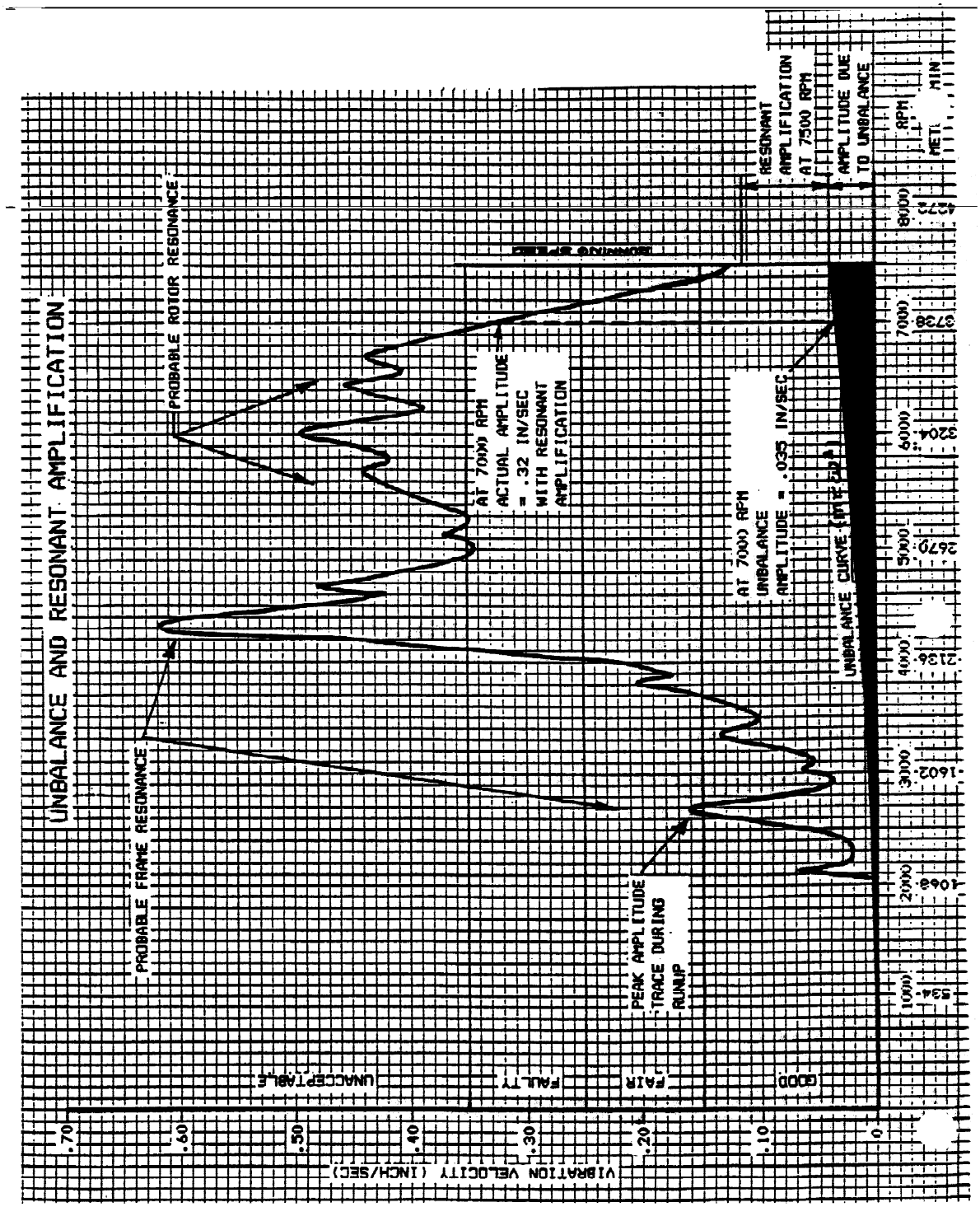

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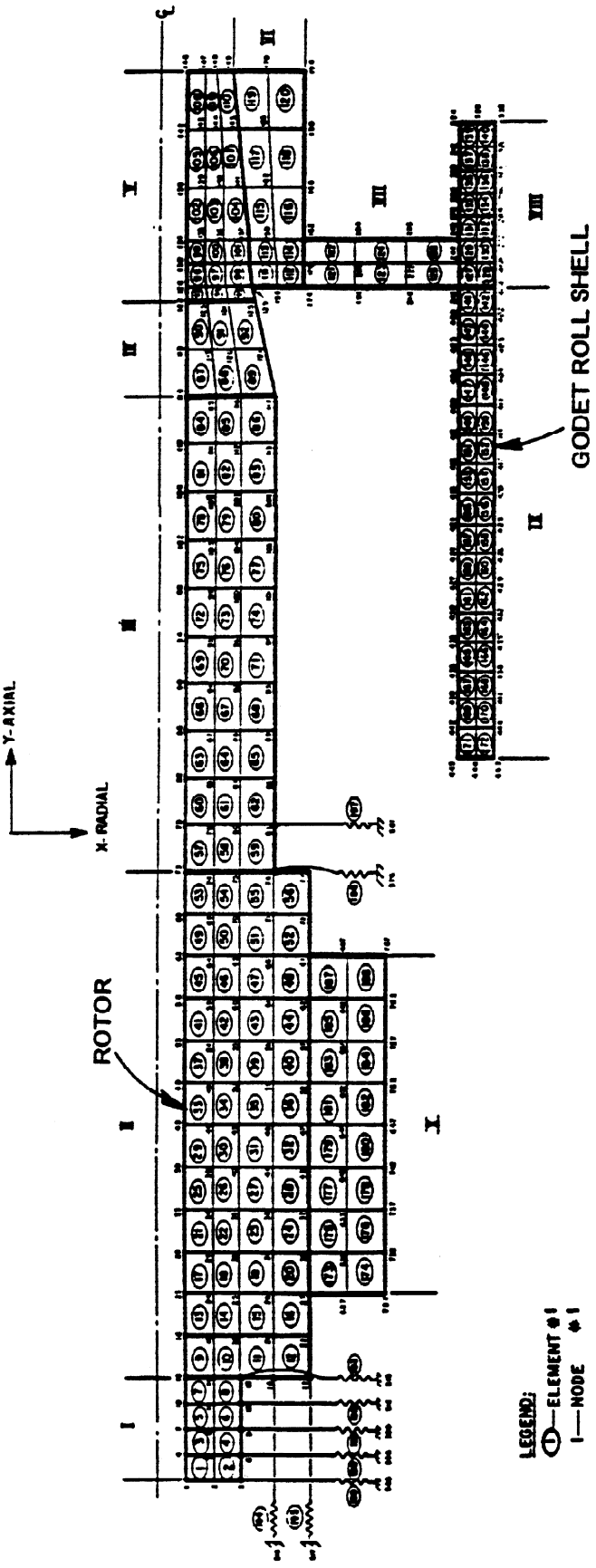

FIGURE 5.
FINITE ELEMENT MODEL

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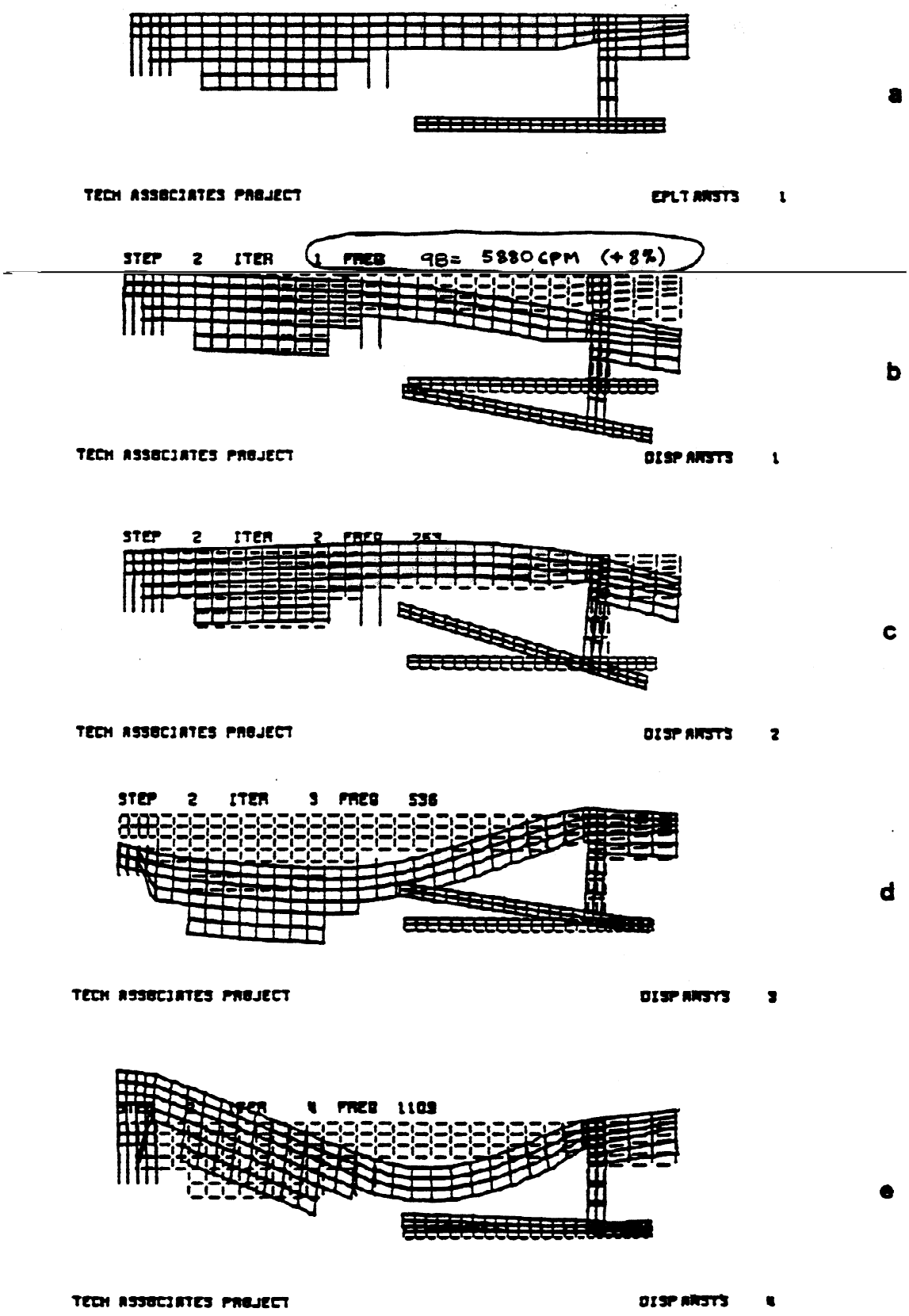

FIGURE 6.
MODE SHAPES OF ORIGINAL SHAFT AND ROLL

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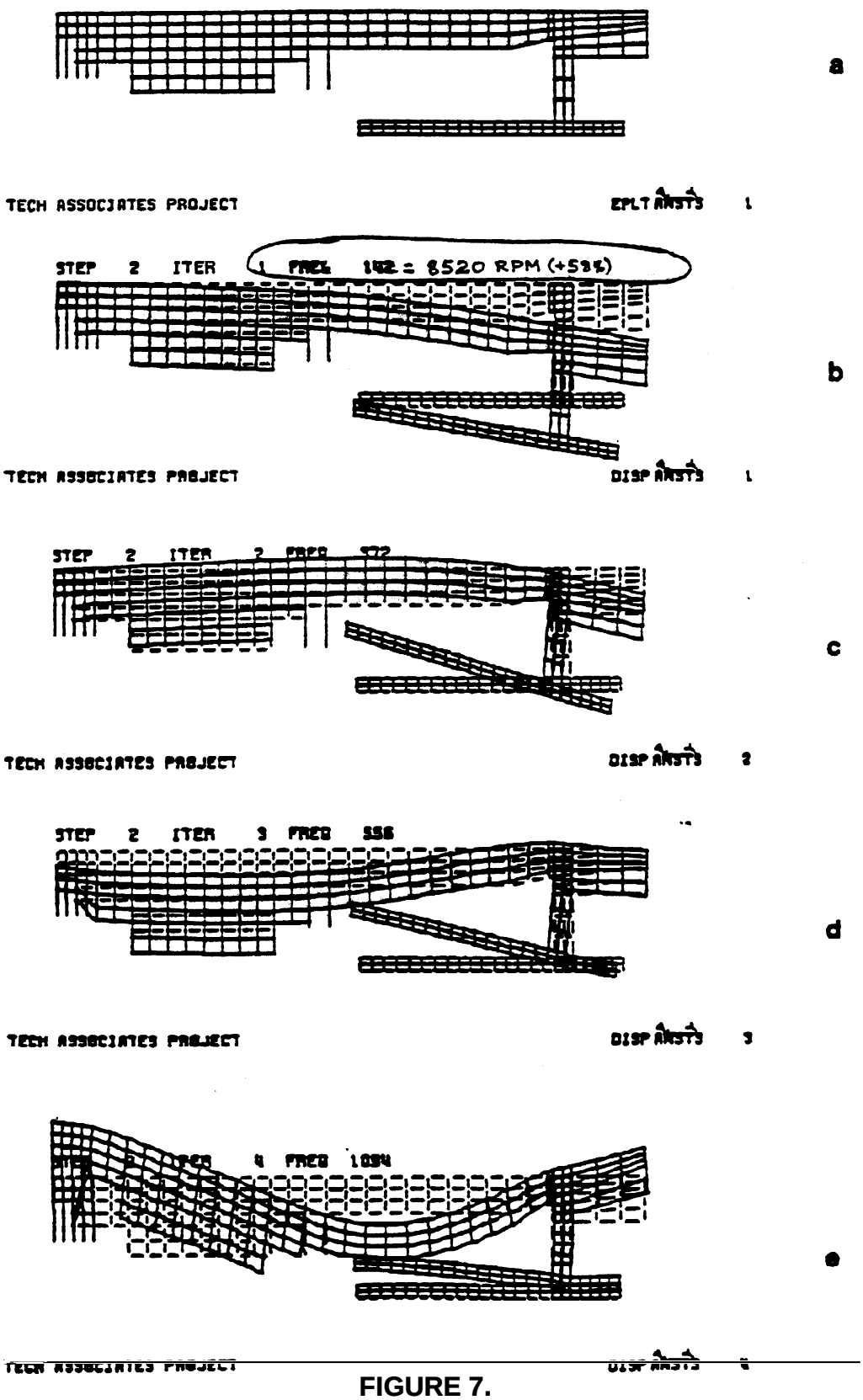

FIGURE 7.
MODE SHAPES OF ORIGINAL SHAFT AND ALUMINUM ROLL

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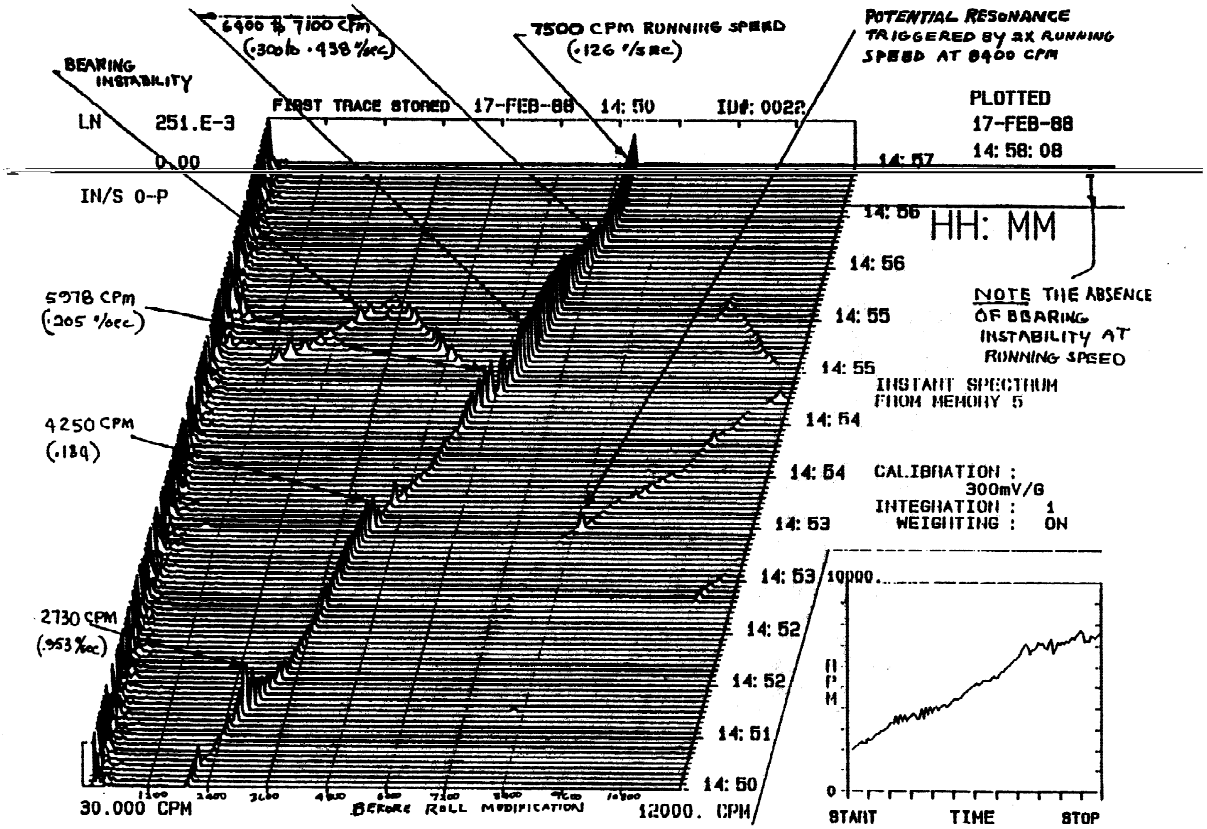

FIGURE 3.

CONTROLLED RUNUP TEST FOR GODET ROLL 3A

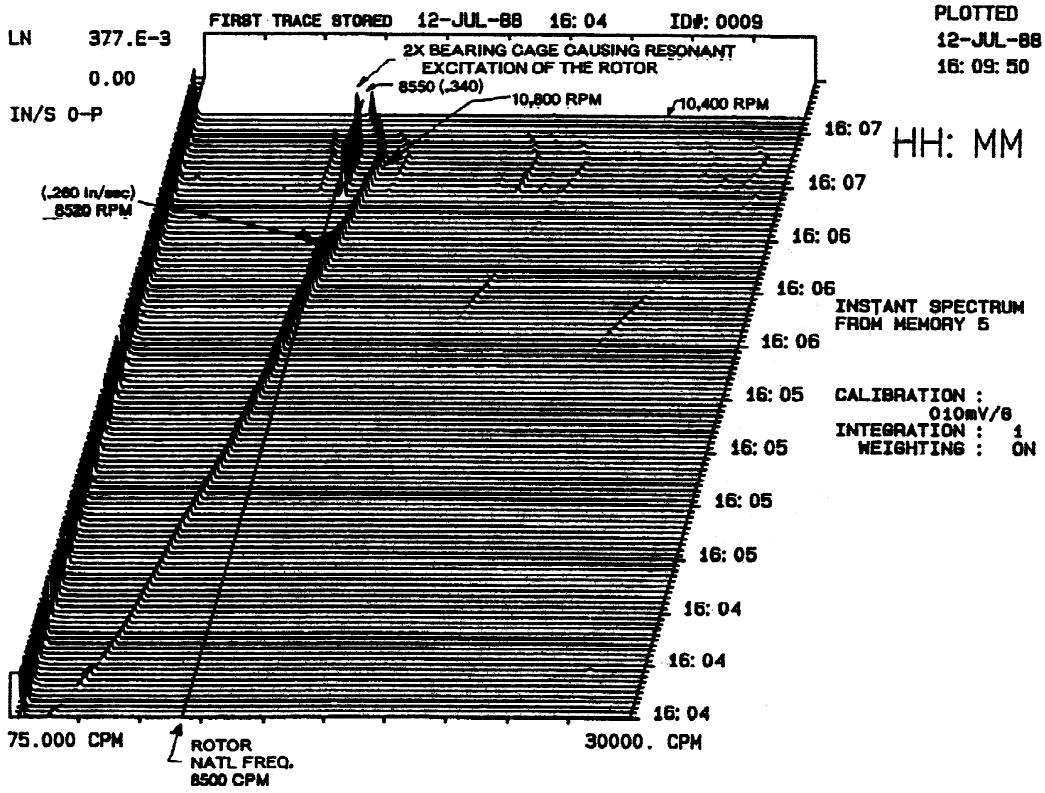

FIGURE 8.

CONTROLLED RUNUP TEST AFTER MODIFICATIONS TO GODET ROLL 3A

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CORRECTION OF INCINERATOR FAN WITH A LOOSE BASE AND UNBALANCE (LOOSENESS TYPE A - 1X RPM)

I. BACKGROUND

A client's Predictive Maintenance Program measurements detected high vibration levels at their McGill incinerator fan shown in Figure 1. They diagnosed the problem as fan unbalance and requested that Technical Associates "balance" the fan. The customer did not have balancing capabilities and previous attempts by other outside vendors had not produced satisfactory results. The fan was integrally mounted, overhung fashion, onto the motor shaft as illustrated in Figure 1. The motor rotated at 3574 RPM.

II. RESULTS

Vibration spectral and phase diagnostics were performed before making any balance attempts. The resulting vibration measurements showed the overall vertical vibration at Position 2V had the highest amplitude at .452 in/sec as can be seen in the "BEFORE" column of Table I. Looking at Figure 1 and Table II, vertical phase measurements at Positions 1, 3, 4 and 5 revealed in-phase motion between Positions 1 and 3, in-phase motion between Positions 3 and 4, but a major phase change of 145° between Positions 4 and 5 (that is, between the motor baseplate and the concrete foundation). The horizontal overall vibration at Position 2H was only .228 in/sec. Spectra for motor Positions 2H and 2V were dominated by 1X RPM vibration as seen in the "BEFORE" spectra of Figures 2A and 3A indicating the likelihood of unbalance existed (.028 in/sec at 2H while 2V had .430 in/sec at 1X RPM). However, since a vertical vibration much higher than a horizontal vibration at 1X RPM is often indicative of a loose/weak/broken base, and because of the vertical phase anomaly, a visual inspection was performed looking for this possibility. The inspection found the anchor bolt nut at Position 6 on Figure 1 was rusted (frozen) onto the anchor bolt threads in a "backed off" condition. Thus at Position 6, the base was able to vibrate freely in a vertical direction. After the nut was freed by heating, it was tightened securely against the base flange. Figures 4A and 4B show the 1X RPM vertical vibration at Position 3 on the motor base where 1X RPM vibration fell from .255 in/sec to .108 in/sec "AFTER" tightening (a 58% reduction before any balancing was performed). This was a significant improvement. Without tightening this nut, the ensuing balance would likely have been difficult and unfruitful.

The fan balance then proceeded in a normal fashion from this point on, reducing vibration levels as noted in the "AFTER" column of Table I. Position 2H overall vibration decreased to .099 in/sec from .228 in/sec and the 2V vibration to .144 in/sec from .452 in/sec. The "AFTER" balancing spectra for Positions 2H and 2V are shown in Figures 2B and 3B, respectively. Notice in the Position 2H and 2V spectra of Figures 2B and 3B that little effect was seen at 7200 CPM as a result of the bolt tightening and fan wheel balance. This was due to the 7200 CPM frequency peak being composed of both a 2X RPM component at 7148.7 CPM and a 2X electrical line frequency (2X 3600 CPM) component at 7200 CPM as seen in the "zoom" spectra of Figure 5 taken on Position 2V before the bolt tightening and balance. Therefore, in this case, tightening loose bolts and balancing did not lower vibration from an electrical source at 7200 CPM.

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III. CONCLUSIONS

A. The “unbalance” problem was actually a combination of looseness and unbalance.

B. Though the “unbalance” vibration appeared to lessen when an anchor bolt was tightened, it was actually the looseness that was being reduced, and not the unbalance.

C. Tightening bolts and balancing did not correct slight electrical problems.

IV. RECOMMENDATIONS

A. Investigate all “unbalance” problems prior to any balance attempts. Often, additional conditions will be detected that must be corrected prior to balancing. Balancing is usually the last corrective measure taken.

B. It was also recommended that the slight electrical problem likely due to slight stator problems be trended in future spectra taken on this machine.

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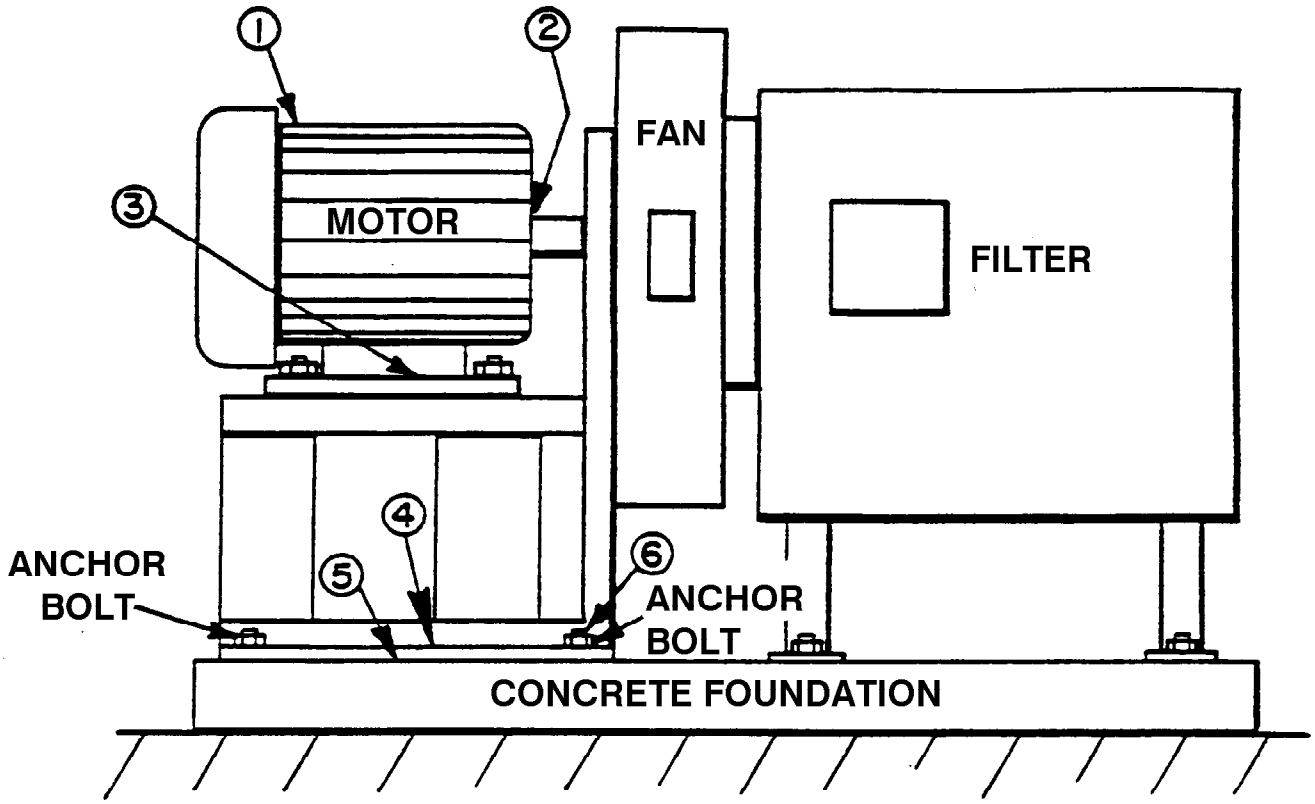

FIGURE 1

TABLE I. AMPLITUDES BEFORE AND AFTER BALANCING INCINERATOR FAN

POSITION	PHASE MEAS.	OVERALL (IN/SEC)			1 X RPM (IN/SEC)		
		BEFORE	AFTER	% CHANGE	BEFORE	AFTER	% CHANGE
1A	20°	.193	.140	- 27%	.155	.090	- 42%
1H	5°	.410	.196	- 52%	.392	.172	- 56%
1V	85°	.203	.137	- 33%	.145	.072	- 50%
2A	30°	.324	.148	- 54%	.306	.126	- 59%
2H	10°	.228	.099	- 57%	.208	.057	- 73%
2V	95°	.452	.144	- 68%	.430	.103	- 76%

TABLE II. VERTICAL PHASE ANALYSIS BEFORE CORRECTIVE ACTIONS

COMPONENT DESCRIPTION	POS. NO.	VERTICAL PHASE	EVALUATION
Top of Motor	1V	85°	Reference Phase
Motor Support Frame	3V	95°	OK (In-Phase)
Motor Baseplate	4V	110°	OK (In-Phase)
Concrete Foundation	5V	255°	145° Phase Change (Excessive)

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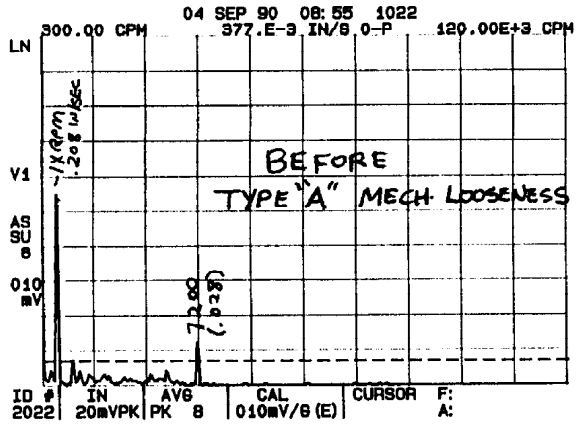


IN/S 0-P	CPM
208.E-3 -1X	3600.0
27.5E-3	7200.0
46.5E-3	35700.

PEAK PICK ON TRACE 1

O/A .228 in/sec

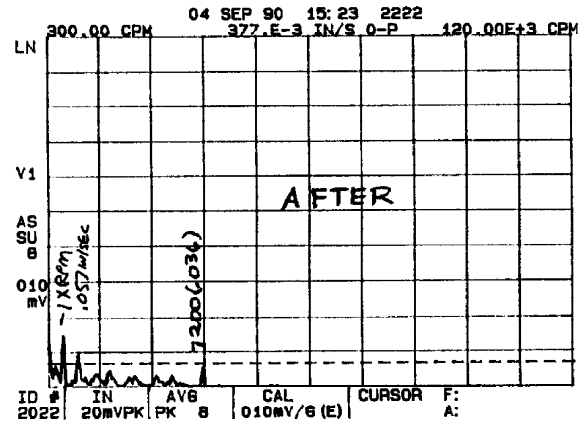
POS. 2 HOR-BEFORE TIGHTENING BOLTS AND BALANCING

05-SEP-90

09: 38: 08

FIGURE 2A

MCGILL FAN



IN/S 0-P	CPM
56.8E-3 -1X	3600.0
36.1E-3	7200.0

PEAK PICK ON TRACE 1

O/A .099 in/sec

POS. 2 HOR-AFTER TIGHTENING BOLTS AND BALANCING

05-SEP-90

09: 58: 58

FIGURE 2B

MCGILL FAN

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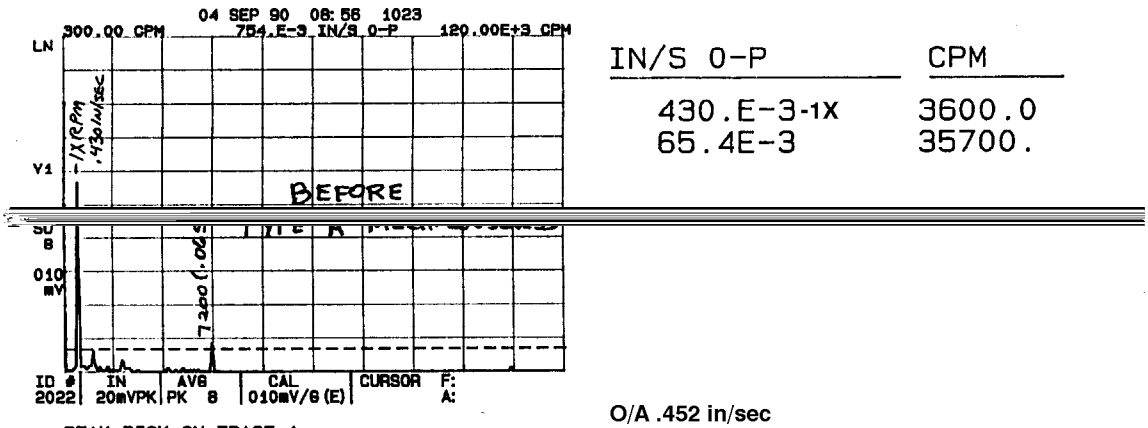


IN/S 0-P	CPM
430.E-3-1X	3600.0
65.4E-3	35700.

O/A .452 in/sec

PEAK PICK ON TRACE 1

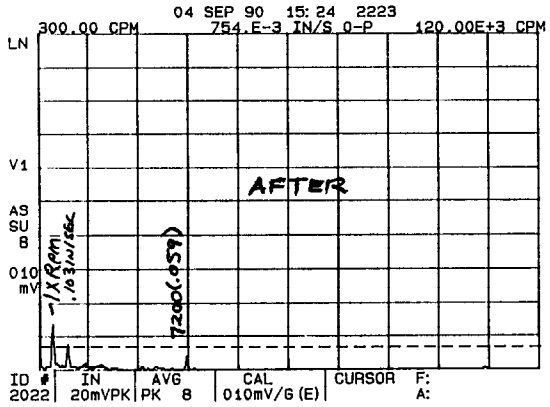
POS 2 VERT-BEFORE TIGHTENING BOLTS AND BALANCING

05-SEP-90

09: 39: 25

FIGURE 3A

MCGILL FAN



IN/S 0-P	CPM
103.E-3-1X	3600.0
58.5E-3	7200.0

O/A .144 in/sec

PEAK PICK ON TRACE 1

POS .2 VERT-AFTER TIGHTENING BOLTS AND BALANCING

05-SEP-90

10: 42: 51

FIGURE 3B

MCGILL FAN

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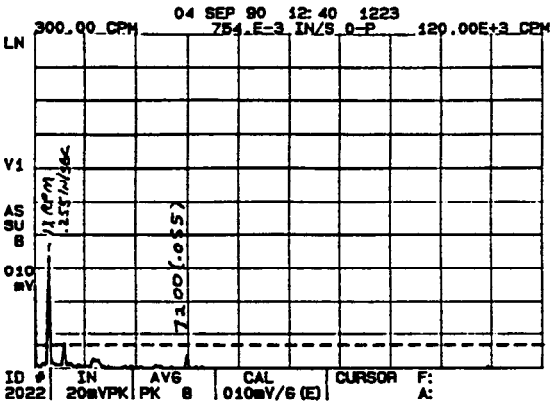


IN/S 0-P	CPM
255.E-3-1X	3600.0
55.1E-3	7200.0

PEAK PICK ON TRACE 1

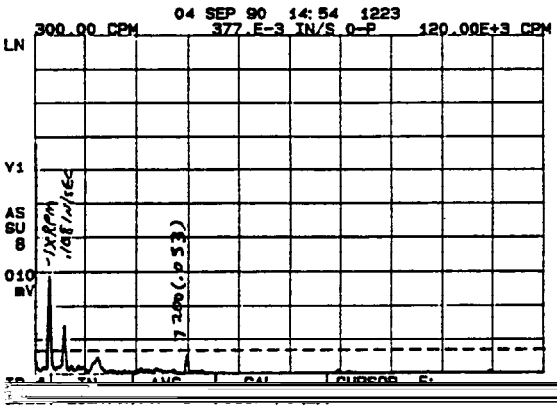
O/A .275 in/sec
POS 3 VERT-BEFORE BOLTS TIGHTENED AND BEFORE BALANCING

05-SEP-90

09: 50: 25

FIGURE 4A

Position 3-V
MCGILL FAN



IN/S 0-P	CPM
108.E-3-1X	3600.0
53.4E-3	7200.0

PEAK PICK ON TRACE 1

O/A .135 in/sec
POS 3 VERT-AFTER BOLTS TIGHTENED AND BEFORE BALANCING

05-SEP-90

09: 51: 36

FIGURE 4B

Position 3-V
MCGILL FAN

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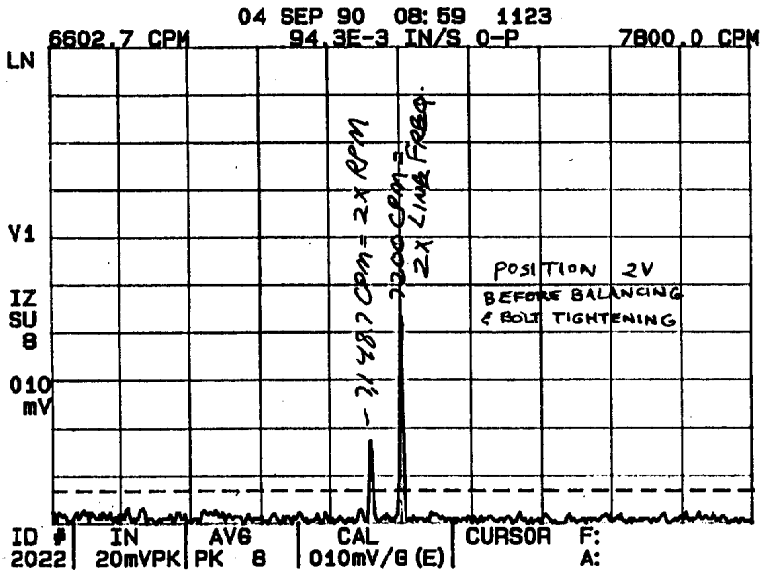

PEAK PICK ON TRACE 1

IN/S 0-P		CPM
16.4E-3	2X RPM	7148.7
40.9E-3	2X ELECTRICAL LINE FREQ. (120 HERTZ)	7200.0

05-SEP-90

09:41:51

POSITION 2-V

MCGILL FAN

FIGURE 5
"ZOOM" SPECTRUM AT POSITION 2V SEPARATING 2X RPM (7,148.7 CPM) FROM 2X ELECTRICAL LINE FREQUENCY (2X 3600 CPM = 7200 CPM)

UNBALANCE AND LOOSENESS PROBLEMS ON AN IN-LINE INDUCED DRAFT FAN (LOOSENESS TYPE B FROM DIAGNOSTIC CHART)

I. BACKGROUND

A manufacturer of in-line induced draft fans used in HVAC ducts wanted to see if vibration levels in their production models could be reduced through balancing. The fan had a small paddle wheel impeller mounted directly on the shaft of a fractional horsepower motor in the configuration illustrated in Figure 1. The unit is mounted to duct work when in use.

II. RESULTS

The initial fan vibration spectra showed an unbalance at 1X RPM of .250 in/sec along with .052 in/sec of looseness at 2X RPM as seen in Figure 2. The overall vibration was .256 in/sec at this time. After the addition of 1.2 grams of balance correction weight, the unbalance was successfully reduced to .053 in/sec - a 79% reduction. Figure 3 pictures the “After” spectra. The “After” looseness at 2X RPM was .089 in/sec, approximately the same as before the balance. The overall vibration was now .108 in/sec - a 58% reduction. The balance was a success. The looseness problem manifested at 2X RPM was relatively unaffected.

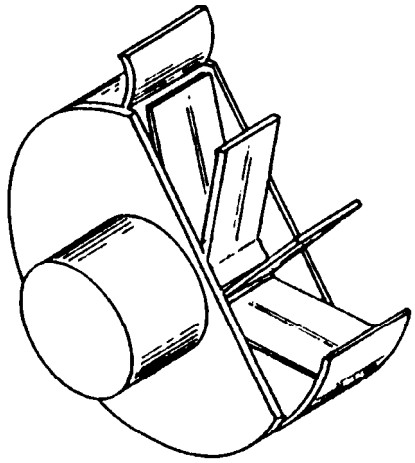

FIGURE 1
IN-LINE INDUCED DRAFT FAN

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III. CONCLUSIONS

A. Correcting unbalance does not remove a pre-existing looseness condition, or practically any other position for that matter. Balancing may improve other existing conditions since it reduces the amount of vibration energy but, it does not solve them. Even at the supposed "unbalance" frequency of 1X RPM, other conditions may exist that will limit the improvements achievable through balancing. Balancing is done only after all other conditions (i.e., misalignment, looseness, etc.) are addressed.

B. The overall vibration in this machine did not improve as much as did the 1X RPM vibration. This results from the premise that correcting unbalance does not correct other conditions that also affect the overall vibration levels. Further reduction in the overall vibration level would require correcting the looseness condition.

C. The final balance weight was 1.2 grams, about half the weight of a dime or two No. 1 paper clips. This is an indication of the system's sensitivity. In this case, it may not take long for dirt to build up a 1.2 gram unbalance. Extreme sensitivity often results from looseness, resonance or a weak mounting structure. It is desirable to reduce the sensitivity levels in extremely sensitive machines.

IV. RECOMMENDATIONS

A. It was recommended that the source of the 2X RPM looseness vibration be identified. The client was told to look closely for possible cracks in the frame, loose bolts or possibly even misaligned (cocked) bearings. Correction of these problems would likely greatly reduce vibration at 2X RPM.

B. A balancing stand was recommended for the plant to enable them to balance the fan wheels right off the production line. A balance tolerance of ISO G-2.5 was recommended to provide satisfactory balance with vibration at 1X RPM likely below approximately .075 in/sec peak velocity.

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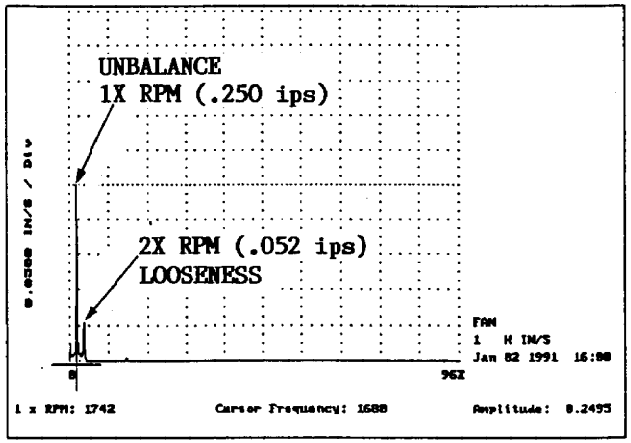


003% x 0.500 = 0.0150 1 x RPM: 1742.00

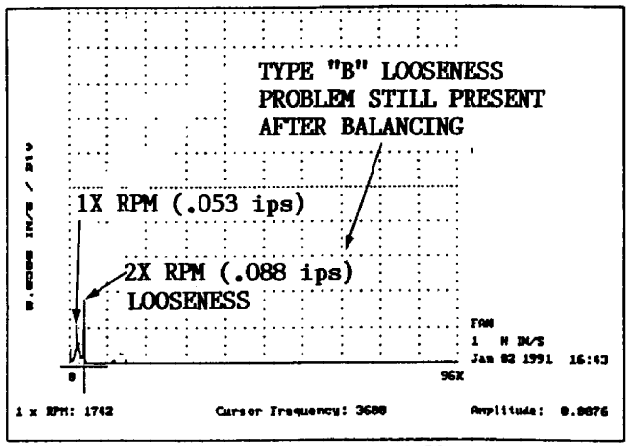
Machine		Pos	Dir	Units	Date
FAN		1	H	IN/S	Jan 02 1991 16:00

Frequency	1 x RPM	Amplitude	
240	0.14	0.0207	
1680	0.96	0.2495	-1X RPM (UNBALANCE)
3600	2.07	0.0516	-2X RPM (LOOSENESS)

BEFORE BALANCING
OVERALL VIBRATION LEVEL .256 in/sec

T - Threshold adjust Q - Quit

FIGURE 2



3% x 0.500 = 0.0150 1 x RPM: 1742.00

Machine		Pos	Dir	Units	Date
FAN		1A	H	IN/S	Jan 02 1991 16:43

Frequency	1 x RPM	Amplitude	
240	0.14	0.0194	
1680	0.96	0.0528	-1X RPM (UNBALANCE)
2400	1.38	0.0279	
3600	2.07	0.0876	-2X RPM (LOOSENESS)

AFTER BALANCING
OVERALL VIBRATION LEVEL .108 in/sec

T - Threshold adjust Q - Quit

FIGURE 3

DETECTION OF FAN WHEEL ROTOR LOOSENESS ON A NUVA FEEDER VENT FAN

I. BACKGROUND

A utility company contracted Technical Associates to establish and install a predictive maintenance vibration program at each of their fossil fuel plants. Additionally, the client wanted Technical Associates to acquire the baseline measurements and issue a **Baseline Report**. This report would include the diagnoses of all problems found ranking these problems in order of severity, a condition evaluation of each machine tested, a drawing of each machine evaluated showing the locations where measurements were acquired and a plant layout showing the location of each machine which had been baselined.

II. DISCUSSION OF RESULTS

There were many machines baselined and many problems were found. One particular problem was found on the Nuva Feeder Vent Fan 68N as seen in the machine sketch Figure 1. The fan had an overall vibration of .460 in/sec (which caused it to exceed its alarm level of .375 in/sec) at Position 4 horizontal as seen in the Machine Report of Figure 2. The Position 4H spectrum, Figure 3, showed some well defined multiple harmonics of running speed with a large 1X RPM amplitude of .340 in/sec. Technical Associates defines this spectral appearance as a Type "C" looseness problem, often attributed to a poor bearing fit on the shaft or in its housing (pillowblock in this case). However, it should be pointed out this same signature can result from a bearing turning on a shaft (a Stage 4 bearing failure condition).

This Type "C" looseness is often the result of a replacement anti-friction bearing being installed on a worn (or undersized) shaft, or in a "wallowed-out" (oversized) bearing housing. A Stage 4 bearing catastrophic failure can occur due to total degradation of the bearing components (i.e., rolling elements and races) finally resulting in an unstable mechanical looseness. Neither the Type "C" nor Stage 4 looseness provide sufficient constraints to keep a shaft on its center of rotation which results in amplification of even a small imbalance.

III. CONCLUSIONS

A. The Position 4 bearing had a mechanical looseness condition that could be a result of bearing fit, bearing wear, bearing turning on a shaft, broken structures and/or loosened fasteners.

B. The unbalance may not have been a true unbalance, but may have resulted from the looseness condition aggravating and amplifying an otherwise insignificant unbalance.

IV. RECOMMENDATIONS

A. Correct the looseness condition.

B. Make a careful field check to ensure neither fan bearing was turning on the shaft.

C. After correcting the looseness condition, re-evaluate the imbalance and balance if necessary.

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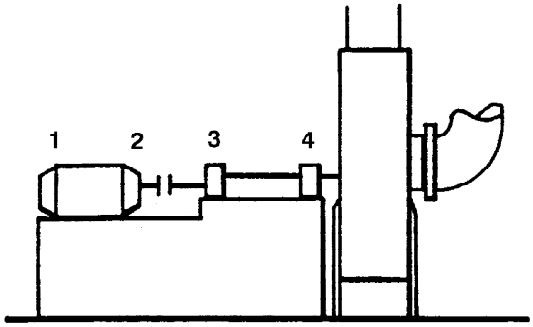
Technical Associates of Charlotte (704 333 9011)

MACHINERY DATA GUIDE SHEET

RELIABILITY SURVEY — predictive maintenance program — (Page__of__)

67 N
67 S
(68 N)
68 S

DETECTION OF LOOSENESS ON A NUVA FEEDER VENT FAN



AREA:_____

MACHINE IDENT: NUVA FEEDER VENT FAN

ROUTE:_____

DRIVER	INTERMEDIATE	DRIVEN
RPM min : 1750	RPM in :	RPM min : 1750
max :	out :	max :
MFGR :	RATIO:	MFGR. :
MODEL :	MODEL :	MODEL :
SER NO. :	SER NO. :	SER NO. :
V___ P___ A___	RATING :	RATING :
SHEAVE : ___DIA.	BELT # :	#BLADES :
BRG#: A (I)	BRG#: (I)	BRG#: A (I)
BRG#: (O)	BRG# (O)	BRG#: (O)
H.P.: 10	GEAR T: CT.	GEAR T: CT.
OTHER:	OTHER:	OTHER:

DCB -17H

FIGURE 1

CF3 R-1092-1

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IRD Mechanalysis Data Analysis System
Dec-04-89

Machine Report (Latest Data) for: NUVA FV FAN 68N

Date	Time	Pos	Dir	Ampl	Units	Change	Alarm	Inspection
Aug-17-89	10:39	1	H	.190	IN/S	------%	.375	
Aug-17-89	10:39	1	H	.042	g/SE	------%	.750	
Aug-17-89	10:39	1	O	.132	IN/S	------%	.375	
Aug-17-89	10:40	1	V	.092	IN/S	------%	.375	
Aug-17-89	10:40	2	H	.130	IN/S	------%	.375	
Aug-17-89	10:40	2	H	.256	g/SE	------%	.750	
Aug-17-89	10:40	2	V	.089	IN/S	------%	.375	
Aug-17-89	10:41	3	A	.309	IN/S	------%	.375	
Aug-17-89	10:41	3	H	.132	IN/S	------%	.375	
Aug-17-89	10:41	3	H	0.72	g/SE	------%	.750	
Aug-17-89	10:41	3	V	.177	IN/S	------%	.375	
Aug-17-89	10:42	4	A	.296	IN/S	------%	.375	
Aug-17-89	10:42	4	H	*.460	IN/S	------%	.375	
Aug-17-89	10:42	4	H	.229	g/SE	------%	.750	
Aug-17-89	10:43	4	V	.240	IN/S	------%	.375	

15 lines this report

FIGURE 2

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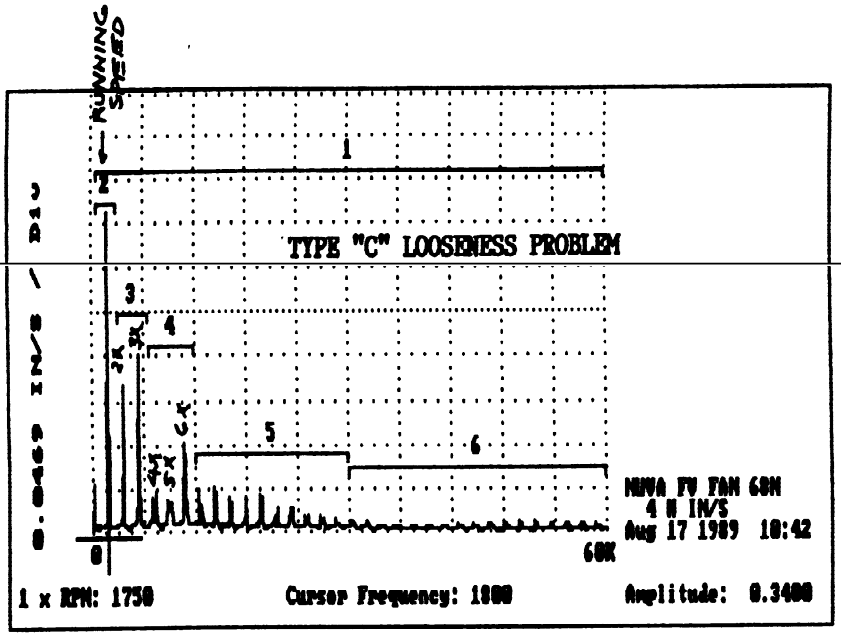

002% x 0.500 = 0.0100 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
NUVA FV FAN 68N	4	H	IN/S	Aug 17 1989 10:42

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
150	0.09	0.0525	19800	11.31	0.0375
1800	1.03	0.3400	21300	12.17	0.0175
3600	2.06	0.1575	21600	12.34	0.0275
5400	3.09	0.1875	23100	13.20	0.0275
6900	3.94	0.0375	23400	13.37	0.0275
7350	4.20	0.0450	24750	14.14	0.0175
8700	4.97	0.0350	25200	14.40	0.0175
9150	5.23	0.0325	26550	15.17	0.0175
10650	6.09	0.0975	27000	15.43	0.0150
12300	7.03	0.0475	28350	16.20	0.0150
12600	7.20	0.0300	30150	17.23	0.0125
14100	8.06	0.0500	30450	17.40	0.0125
14400	8.23	0.0425	32100	18.34	0.0125
15900	9.09	0.0375	48000	27.43	0.0125
16200	9.26	0.0325	49800	28.46	0.0125
17850	10.20	0.0375	51600	29.49	0.0125
19500	11.14	0.0400			

T - Threshold adjust Q - Quit

FIGURE 3

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DIAGNOSIS AND CORRECTION OF A SERIOUS RUB PROBLEM ON A CHEMICAL PRODUCT REACTOR OPERATING AT 5 RPM TO 60 RPM

I. BACKGROUND

Two chemical reactors had just been installed and were experiencing high vibration and unusual noise within a new plant facility. At its worst, the problem actually shook the entire building structure and a noticeable “impact” sound was heard with each revolution of the unit. Discussions prior to making measurements indicated that the main focus should be on the bearing and drive assembly since a vessel inspection had shown no internal clearance problems.

II. ANALYSIS RESULTS

Figures 1 and 2 include drawings on the chemical reactor evaluated. Figure 1 shows the hydraulic drive assembly which included a shaft of roughly 8 inch diameter. Figure 2 shows the double helix blading that was driven by the hydraulic unit (note that it also shows each of the two bearings in phantom). Note the great overhang below the bearings for this reactor blading and shaft which is oriented vertically. Note that there was no bearing support beneath the bearings.

A series of vibration measurements were conducted with the reactor running at speeds between 5 RPM and 60 RPM. Some tests were run without any liquid within the reactor. Later tests were conducted initially with water at room temperature, and later with it at higher temperatures. Initially, measurements were made at room temperature with the agitator rotating at only 9 RPM and empty. Plot 1 shows the results of this 9 RPM measurement. The 9 RPM running speed peak was well defined along with many running speed harmonics in the spectrum. During the entire test, it was observed that there was a tendency towards amplification of the harmonics around the vicinity or just above 300 CPM. Plot 2 shows this clearly which is an indication of resonant amplification.

During the first testing, it was difficult to diagnose whether the problem was mechanical looseness or a rub. The many harmonics shown in Plots 1 and 2 are indicative of either problem. However, a test was conducted late in the afternoon which showed clear indication of a rub rather than mechanical looseness. Plot 3 shows this rub indication. Running at 34.5 RPM, the spectrum shows not only running speed and harmonics, but also a whole series of one-half harmonics which are highlighted on the spectrum by dots. Therefore, with the simultaneous presence of running speed accompanied by one-half running speed harmonics, along with the amplification in the area of probable resonance, this was a strong indication of rub. At the very low speed of 9 RPM, the rub was not audible, but clearly showed on the spectrum. Importantly, for this reason, it was pointed out that it must not be assumed that a rub does not exist if it cannot be heard.

Plot 4, taken with the reactor shut down, was captured with the purpose of determining the agitator assembly natural frequency. In this case, a nearby reciprocating chiller was started up and used as the forcing input to an impulse natural frequency test. In this case, the predominant response was at 300 CPM, the suspected resonant area which would be excited in a rub of 2 adjacent components.

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III. RECOMMENDATIONS

Due to the strong indications of possible rub, it was recommended that vendor personnel be sent down into the reactor to make a careful inspection of the double helix blading surfaces as well as the stainless steel tank wall. If the rub location could be confirmed, it was recommended that material be removed as required.

IV. CORRECTIVE ACTIONS TAKEN

Upon subsequent inspection of the reactor vessel on May 19, the vendor representative confirmed a significant rub between the agitator and the vessel wall. The area most effected was down towards the right-hand side of the conical section shown in Figure 2. Therefore, a decision was made to increase the clearance between the blading and vessel wall at locations where it was inadequate. Material was removed from the agitator blading by use of a flame cutting process and was dressed out to a smooth finish by grinding. Approximately a .25 inch width was removed all the way down each of the two helix blades of each reactor. This would therefore increase the clearance between the blading and vessel wall to approximately .625 inch.

V. RESULTS OF CORRECTIVE ACTIONS

On May 30, vibration measurements were conducted by Technical Associates on the reactor to assess the effect of the "grinding". The effect was dramatically illustrated by comparing "Before" and "After" spectra and by comparing "Before" and "After" overall vibration levels. Plots 5 and 6 compare "Before" and "After" spectra. The upper figure of Plot 5 is a spectrum from measurements of May 18 with the reactor running at 9 RPM taken before any grinding was done. The May 30 plot, produced from measurements taken after grinding, is shown by the lower figure of Plot 5. A clean once per revolution signal is shown at 9 CPM with no harmonics. Also, note that the overall vibration dropped from .065 in/sec down to .023 in/sec.

Recall that during heavy rubbing with the unit running at higher speeds, the agitator natural frequency was excited at a frequency of approximately 300 CPM. The spectra in Plot 6 were taken at the same position, but at a speed of 45 RPM. The upper figure shows vibration before grinding and had an excessive amplitude of .603 in/sec at the natural frequency of the agitator. Note that this plot was set up for a full-scale amplitude of 1.10 in/sec in order to keep the natural frequency within scale. The lower plot, an "After" grinding measurement spectrum, shows a resonant amplitude of only .0199 in/sec, a 30 times reduction in vibration. Note the full-scale amplitude setting of only .041 in/sec in this lower plot. A whole series of tests were conducted on May 30 to thoroughly evaluate the reactor. At no time during any of the testing did a serious rub occur. Instead, there was evidence only of small "burrs" of material periodically wedging between the blading and wall. Even when this did occur, the lower spectrum of Plot 6 shows that peak amplitudes were only 3% of those before agitator material removal. Finally, Figure 3 compares overall vibration versus agitator RPM before and after removal of the blading material and shows the dramatic vibration reduction achieved.

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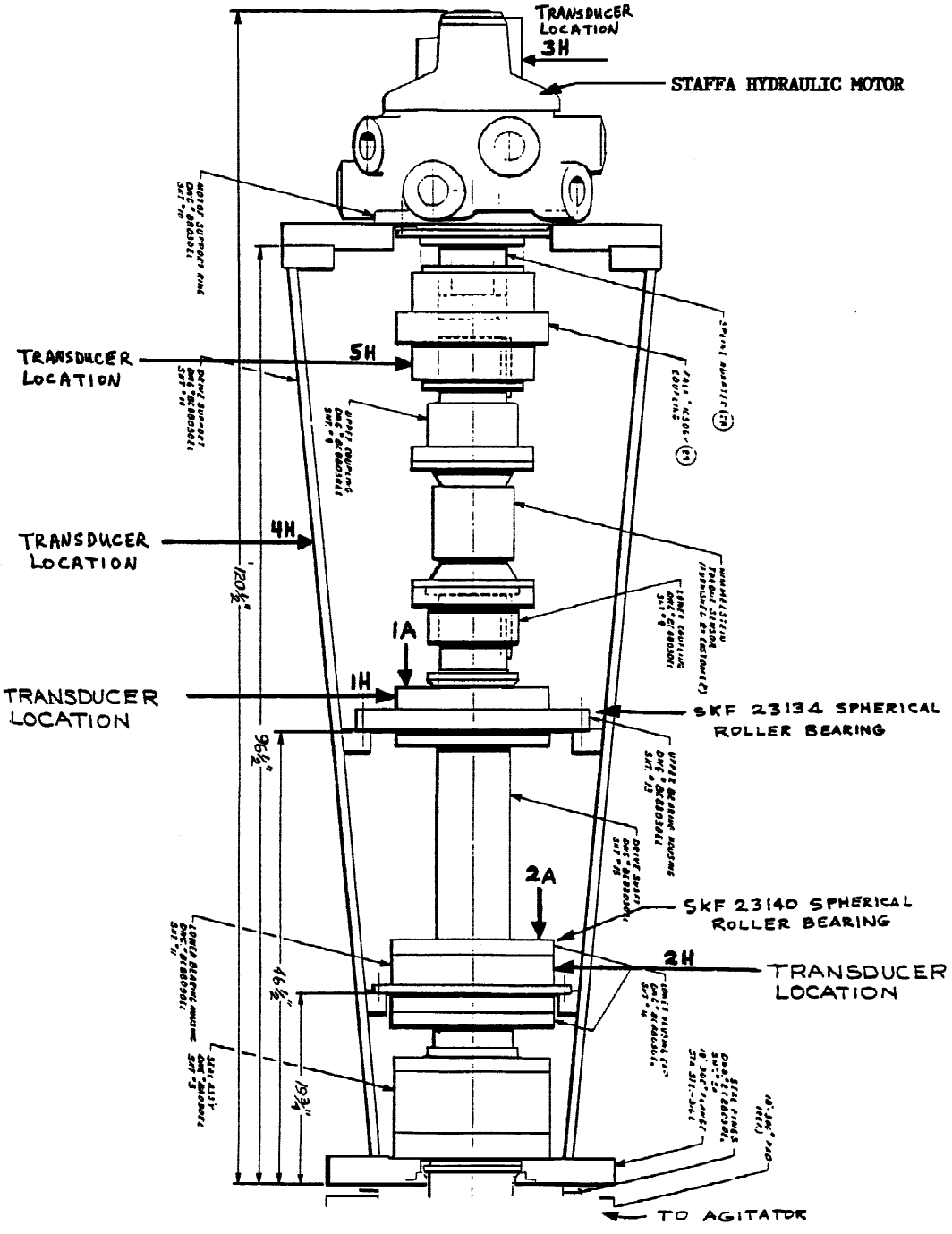

FIGURE 1
AGITATOR DRIVE

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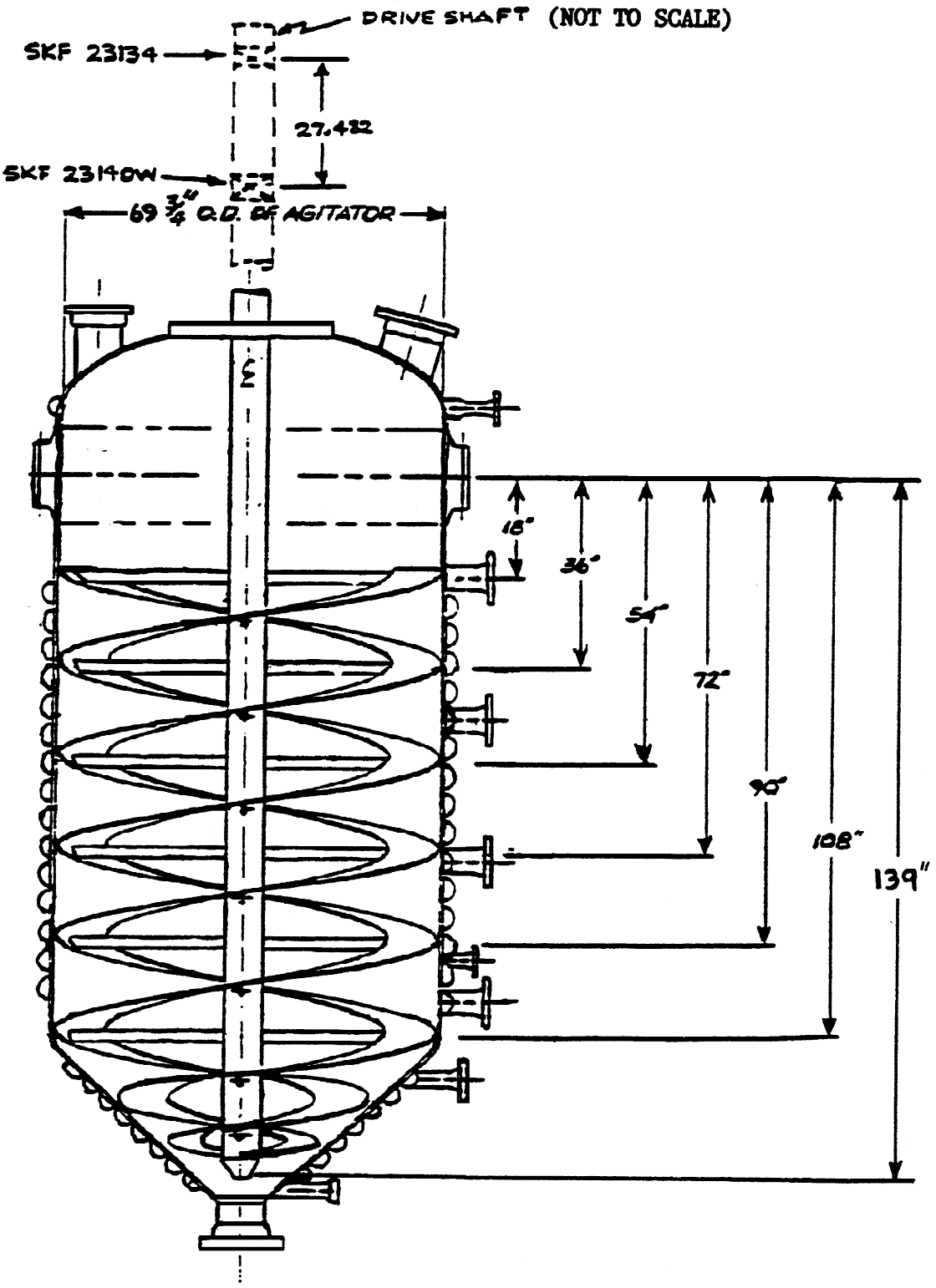

FIGURE 2
AGITATOR

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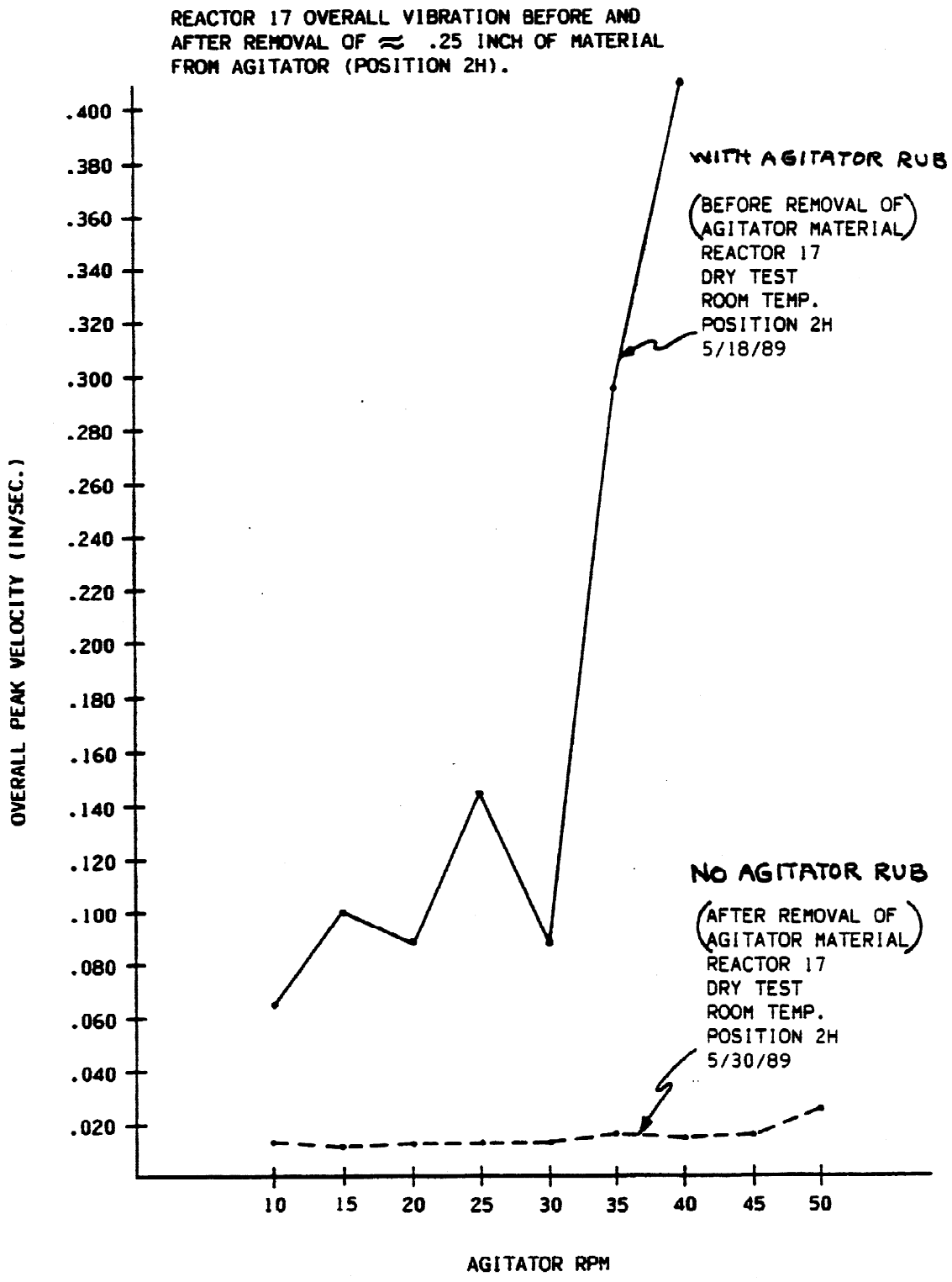

FIGURE 3

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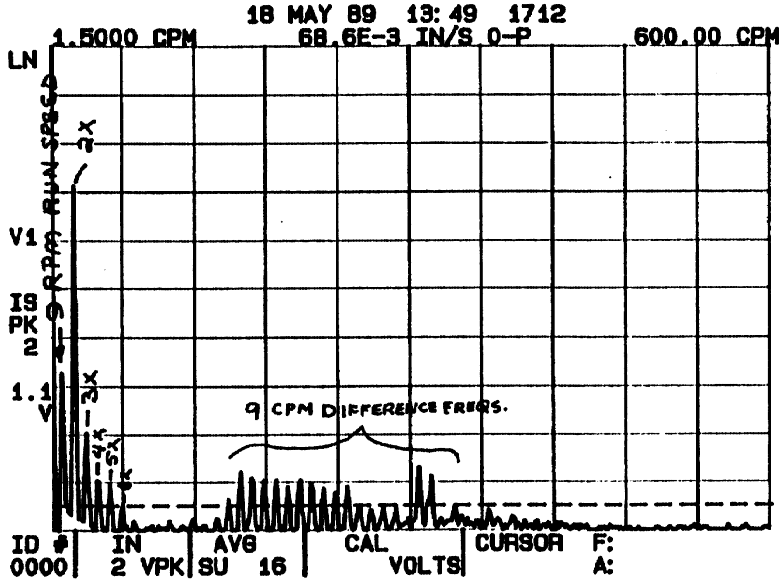

PEAK PICK ON TRACE 1

OA = .065 "/SEC

POSITION 1H

ROOM TEMP.

REACTOR EMPTY

24-MAY-89

11:21:17

VECTRA REACTOR 17

VIBRATION SPECTRUM

RUNNING AT 9 RPM

IN/S 0-P	CPM
22.6E-3	9.0000
49.2E-3	19.500
14.1E-3	30.000
7.20E-3	39.000
6.58E-3	49.500
3.45E-3	60.000
4.39E-3	148.50
8.46E-3	159.00
7.52E-3	167.99
7.20E-3	178.50
7.20E-3	188.99
6.26E-3	198.00
7.20E-3	208.49
6.58E-3	217.50
5.95E-3	227.99
5.33E-3	236.99
6.26E-3	247.50
3.45E-3	288.00
9.08E-3	307.50
4.07E-3	315.00
7.83E-3	318.00

PLOT 1

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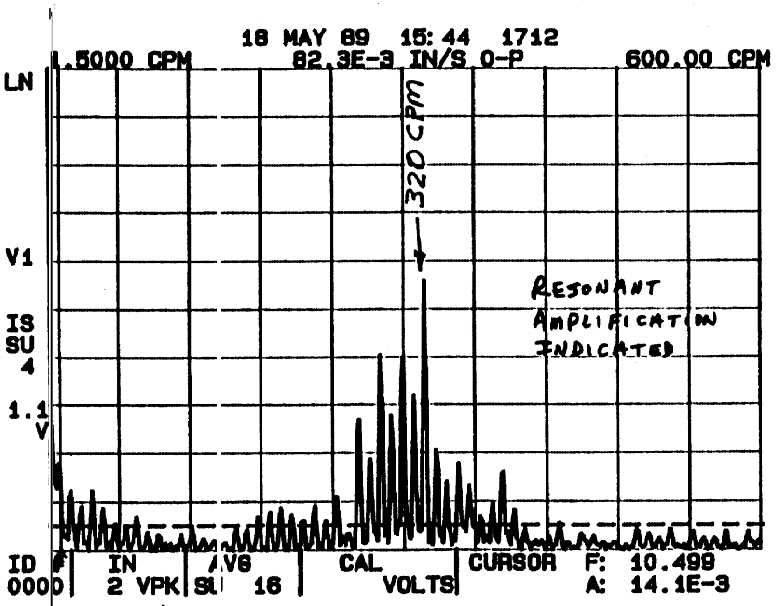

24-MAY-89

11:59:07

VECTRA REACTOR 17
VIBRATION SPECTRUM
RUNNING AT > 9 RPM

IN/S 0-P	CPM
14.7E-3	9.0000
10.1E-3	19.500
7.52E-3	28.500
10.1E-3	37.500
7.14E-3	46.500
4.51E-3	57.000
4.13E-3	66.000
5.64E-3	75.000
4.13E-3	121.50
5.64E-3	176.99
6.39E-3	187.50
7.14E-3	196.49
6.01E-3	205.50
4.89E-3	214.50
7.52E-3	224.99
4.89E-3	234.00
9.02E-3	243.00
22.2E-3	262.50
15.4E-3	271.50
33.4E-3	280.50
6.76E-3	285.00

PLOT 2

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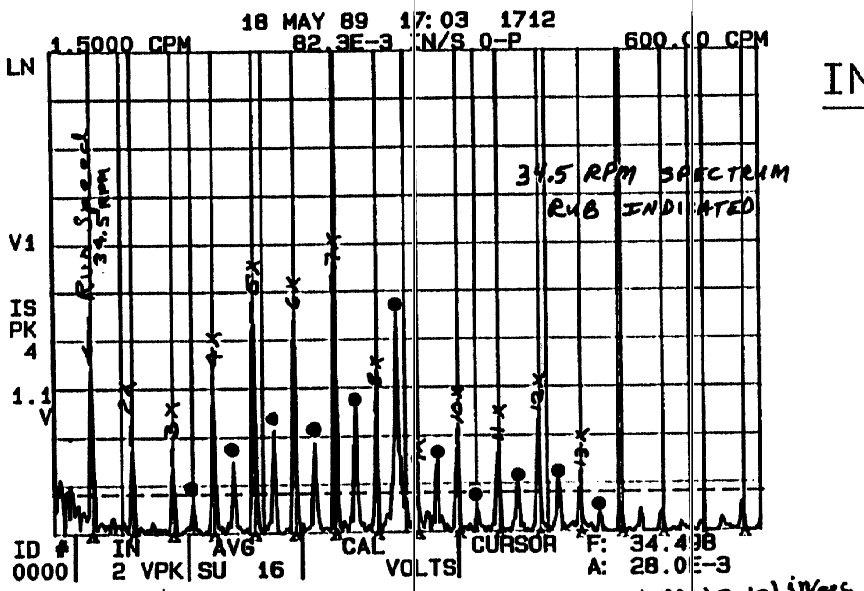

PEAK PICK ON TRACE 1

21-MAY-89

15: 30: 14

VECTRA REACTOR 17

SPECTRUM SHOWING A RUB

● MARKS THE "1/2" HARMONICS

IN/S 0-P	CPM
7.52E-3	4.5000
9.40E-3	7.5000
7.52E-3	13.500
8.27E-3	16.500
27.8E-3	34.500
14.3E-3	69.000
11.3E-3	103.50
22.5E-3	138.00
12.4E-3	154.50
35.7E-3	172.49
17.7E-3	188.99
33.1E-3	205.50
15.4E-3	223.50
39.1E-3	240.00
21.8E-3	258.00
19.2E-3	274.50
38.0E-3	292.50
12.4E-3	327.00
14.3E-3	343.48
13.5E-3	378.00
8.27E-3	394.50

PLOT 3

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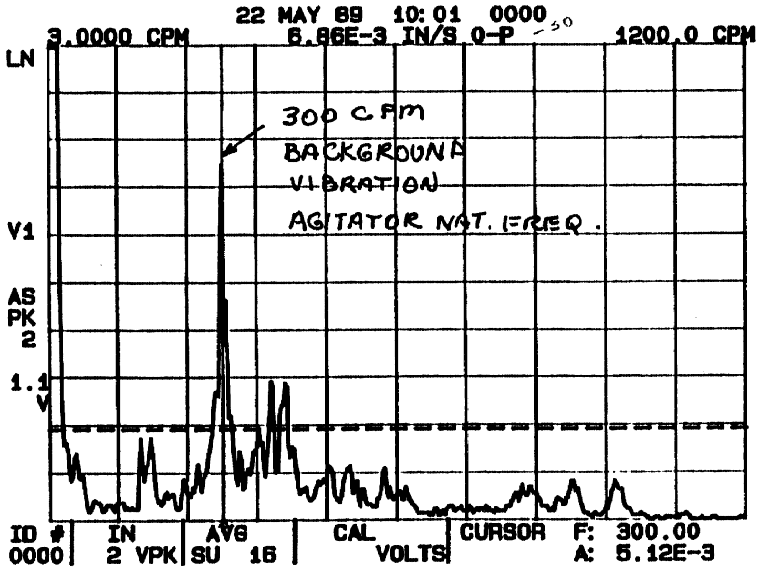


PEAK PICK ON TRACE 1

IN/S 0-P	CPM
1.85E-3	288.00
5.14E-3	300.00
3.20E-3	306.00
1.50E-3	315.00
1.32E-3	363.00
2.00E-3	383.98
1.97E-3	407.98

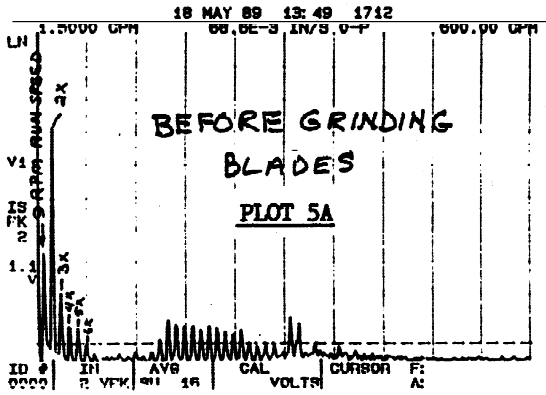
24-MAY-89

12: 13: 46

SHOWING AGITATOR NATURAL FREQUENCY

VEC REA 16

PLOT 4



PEAK PICK ON TRACE 1

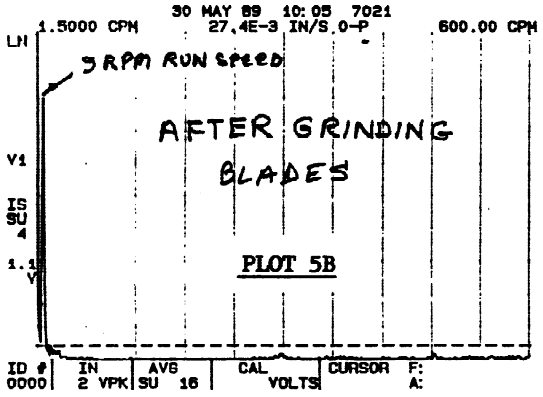
OA = .065 "/SEC

24-MAY-89

BEFORE GRINDING

VECTRA REACTOR 17
VIBRATION SPECTRUM
RUNNING AT 9 RPM

IN/S 0-P	CPM
22.6E-3	9.0000
49.2E-3	19.500
14.1E-3	30.000
7.20E-3	39.000
6.58E-3	49.500
3.45E-3	60.000
4.39E-3	148.50
8.46E-3	159.00
7.52E-3	167.99
7.20E-3	178.50
7.20E-3	188.99
6.26E-3	198.00
7.20E-3	208.49
6.58E-3	217.50
5.95E-3	227.99



PEAK PICK ON TRACE 1

31-MAY-89

OA = .023 "/SEC

AFTER GRINDING

09: 41: 41

VECTRA REACTOR 17
VIBRATION SPECTRUM
RUNNING AT 9 RPM

IN/S 0-P	CPM
22.0E-3	9.0000 RPM
1.25E-3	15.000
1.13E-3	18.000

VEC REA 17

PLOT 5

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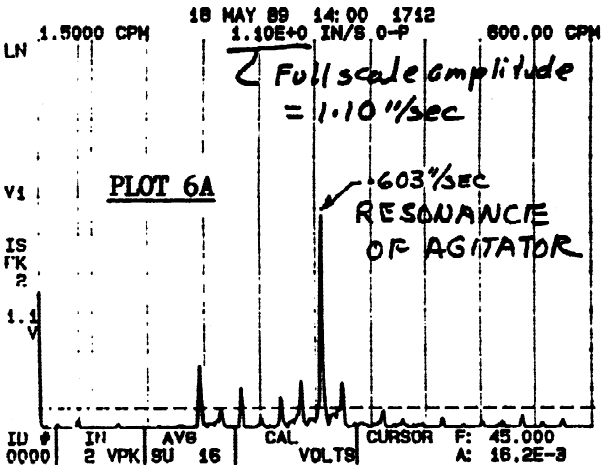
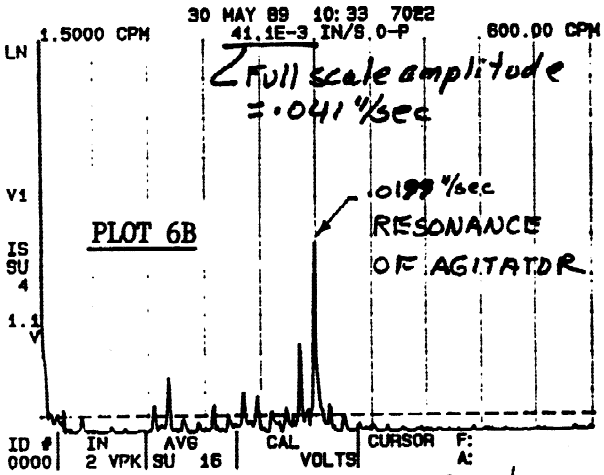


PEAK PICK ON TRACE 1

IN/S 0-P	CPM
176.E-3	175.50
111.E-3	219.00
85.4E-3	262.50
131.E-3	285.00
603.E-3	307.50
55.3E-3	324.00
126.E-3	328.50

OVERALL = .74"/SEC
BEFORE GRINDING
45 RPM

24-MAY-89



PEAK PICK ON TRACE 1

IN/S 0-P	CPM
2.25E-3	10.500
1.88E-3	19.500
1.69E-3	46.500 RPM
2.82E-3	126.00
5.82E-3	141.00
2.82E-3	188.99
1.69E-3	203.99
4.13E-3	220.49
3.75E-3	235.50
2.06E-3	250.50
2.44E-3	267.00
9.20E-3	282.00
2.25E-3	291.00
19.9E-3	298.50
2.82E-3	313.50

OVERALL = .0272"/sec
AFTER GRINDING
45 RPM

31-MAY-89

09: 56: 18

PLOT 6

VEC REA 17

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DETECTION OF ROTOR RUB ON A LEWIS SCREW COMPRESSOR

I. BACKGROUND

During a February 7, 1990 predictive maintenance vibration survey by Technical Associates, a refrigeration screw compressor (Lewis No. 1 RS-05) was found to have developed 1/2X RPM harmonics throughout the motor with the motor axial vibration being the highest. A machine sketch is shown in Figure 1. The motor rotated at a nominal 3560 RPM and was directly coupled to a screw compressor with a 4 lobe male rotor and a 6 lobe female rotor. Due to the presence of the noticeable amplitude peaks at 1/2X RPM, Technical Associates was asked to take a close look at all data to confirm the problem cause and severity as well as to make necessary remedial recommendations.

II. DIAGNOSTIC RESULTS

The spectra, Figures 2 and 3, for the axial motor Positions 1A and 2A illustrate the many 1/2X RPM harmonics that were present in the motor. Notice in Figure 2 not only the presence of many half-harmonics, but the generally raised area in the vicinity of 4.5X RPM. Also note the amplitude at this frequency (16,200 CPM) was an excessive .395 in/sec at a frequency which should not even be present in the spectrum (4.5X RPM).

Unlike the motor spectra, the compressor spectra did not display these 1/2X RPM harmonics. Due to this fact, it was originally thought there may have been a rub of the motor rotor on its stator and/or that problems were present in the coupling. The surprise came after the report was issued (see Figure 4 for the routine PMP report), and the unit was inspected. During this inspection, the rub was found in the compressor in the form of a rotor contacting the compressor's slide valve as reported by the client. Figure 5 shows the "BEFORE" and "AFTER" corrections spectra of Position 1A. The "AFTER" spectra shows no evidence of the 1/2X RPM harmonics.

III. CONCLUSIONS

A. The 1/2X RPM harmonics on the motor resulted from the compressor rotor contacting the compressor's slide valve.

IV. RECOMMENDATIONS

A. Continue the periodic vibration measurements. Look for the possible reappearance of such 1/2X RPM subharmonics in future PMP spectra.

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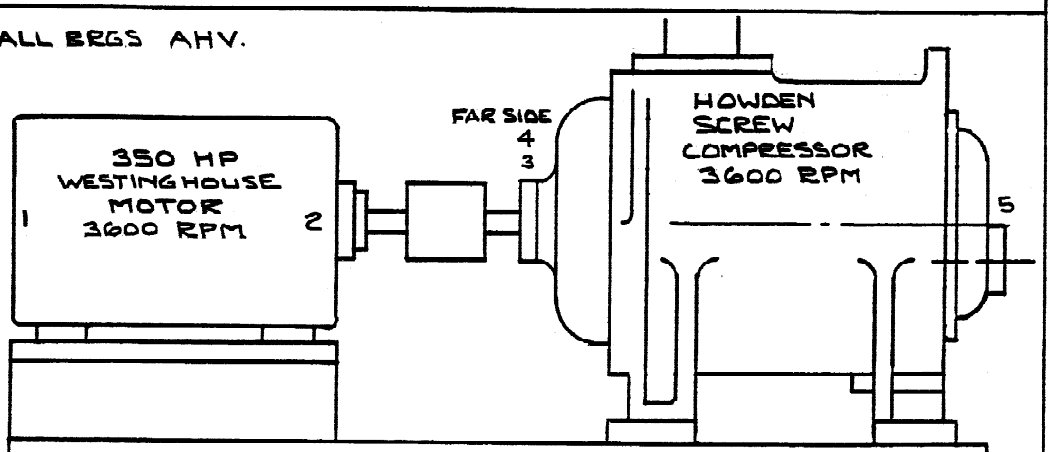
Technical Associates of Charlotte (704-333-9011)

MACHINERY DATA GUIDE SHEET

RELIABILITY SURVEY predictive maintenance program (Page__of__)

ALL BRGS AHV.



DETECTION OF ROTOR RUB ON A LEWIS SCREW COMPRESSOR

COMPUTER = LEWIS #1 RS-05 (#1 COMP)
LEWIS #2 PH-2300 (#2 COMP)

AREA:_____

MACHINE IDENT: LEWIS #1 & #2 REFRIG COMPRESSORS.

ROUTE:_____

DRIVER	INTERMEDIATE	DRIVEN
RPM min : 3560	RPM in :	RPM min :
max :	out :	max :
MFGR WESTINGHOUSE	RATIO:	MFGR. HOWDEN COMP

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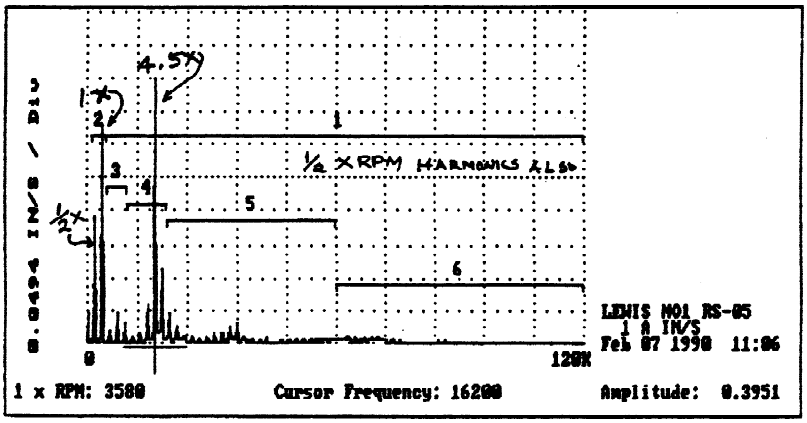

002% x 1.000 = 0.0200 1 x RPM: 3580.00

Machine	Pos	Dir	Units	Date
LEWIS NO1 RS-05	1	A	IN/S	Feb 07 1990 11:06

Frequency	1 x RPM	Amplitude	
300	0.08	0.0450	
1800	0.50	0.1900	½X
3600	1.01	0.3250	1X
5400	1.51	0.0200	1.5X
7200	2.01	0.0450	2X
9000	2.51	0.0300	2.5X
14400	4.02	0.0600	4X
16200	4.53	0.3951	4.5X
18000	5.03	0.1150	5X
19800	5.53	0.0450	5.5X
21600	6.03	0.0249	
34200	9.55	0.0249	
36000	10.06	0.0300	

T - Threshold adjust Q - Quit

FIGURE 2

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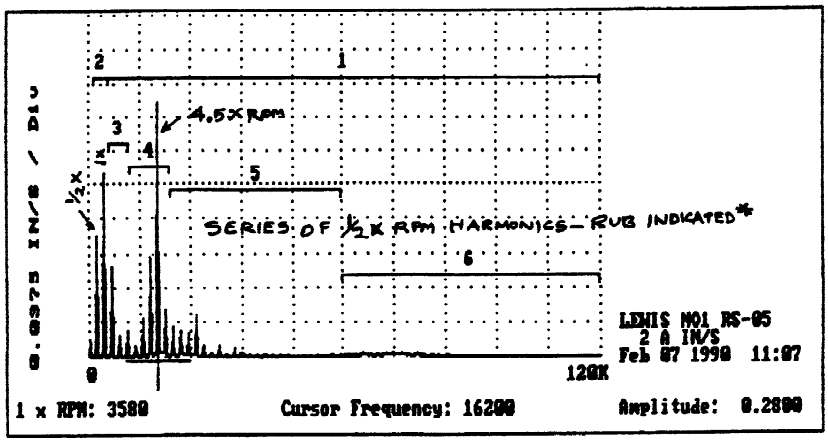

002% x 0.500 = 0.0100 1 x RPM: 3580.00

Machine	Pos	Dir	Units	Date
LEWIS NO1 RS-05	2	A	IN/S	Feb 07 1990 11:07

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
300	0.08	0.0175	34200	9.55	0.0100
1800	0.50	0.1325			
3600	1.01	0.2000			
5400	1.51	0.0975			
7200	2.01	0.0225			
9000	2.51	0.0275			
10800	3.02	0.0125			
12600	3.52	0.0425			
14400	4.02	0.1100			
16200	4.53	0.2800			
18000	5.03	0.0525			
19800	5.53	0.0325			
21600	6.03	0.0275			
23400	6.54	0.0250			
25200	7.04	0.0450			
27000	7.54	0.0125			
30600	8.55	0.0125			

*NOTE: WHEN WENT INTO MACHINE, FOUND ROTOR HAD CONTACTED SLIDE VALVE (ACTUALLY BENT VALVE)

T - Threshold adjust Q - Quit

FIGURE 3

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SUMMARY OF RESULTS AND RECOMMENDATIONS FOR MACHINES FOUND TO HAVE PROBLEMS

(SORTED FIRST BY PROBLEM PRIORITY; THEN BY PROBLEM SEVERITY WITHIN PRIORITY)

AS OF: FEBRUARY 7, 1990

AREA	MACHINE COMPUTER NAME	INDICATED PROBLEM(S): UNBALANCE	MISALIGNMENT/SOFT FOOT	BENT SHAFT	LUBRICATION PROBLEM	BEARING PROBLEM	MECHAN. LOOSENESS	BELT/SHEAVE PROBLEM	RESONANCE PROBLEM	GEAR PROBLEM	BLADE PASSING PROB.	CAVITATION	MOTOR ELECTRICAL	OIL WHIRL /WHIP	LOOSE FOUNDATION	VIB. MONITOR PROB.	SEAL /ROTOR RUB	OTHER (SEE REMARKS)	*PRIORITY OF PROBLEM	HIGHEST OVERALL VIBRATION: POSITION NO.	LEVEL: m = mils, i = in/sec, g = g's	RECOMMENDATIONS & REMARKS	SEE PLOT NO.
	LEWIS NOI RS-05					●	●				●								①	1A	.810	Multiple 1/2 RPM harmonics detected at motor test point locations indicating a relatively severe rub condition. This is probably caused by bearing looseness in the end bells or coupling related problems being transmitted into the motor. Recommend thoroughly inspecting the motor and coupling and repairing as needed. (Note: This should be treated as an Alarm 1 condition.) Multiple harmonics of 4 times RPM were also detected at Position 5A on the compressor. This is due to aerodynamic forces and will be trended in future surveys. This particular type of vibration at lobe bass frequency (4X RPM) on Position 5A is common on these machines (see data guide sheet for this machine to see position numbers).	8 9 10
	HOTWATER PH1232					●													①	3V	.238	Bearing defect frequencies detected on both pump bearings. Recommend replacing both pump bearings.	6 7
	CO2 COMP ME2773							●	●										2	1A	.637	1200 CPM (2 times compressor RPM) is predominant at motor test point locations with Position 1A displaying the maximum amplitude of .595 in/sec (Plot 1). This is probably caused by a belt and/or sheave related problem. Recommend checking belts and sheaves for wear and replacing as needed. Align and tension the belts to specifications. Also recommend checking hold-down bolts on motor base, tightening and/or replacing as required. A loose motor base may be resonant at 2X compressor RPM which would be resolved by addressing the fastener/base problem.	1 2
	FRICK COMP SM55										●								3	6V	.335	Multiple harmonics of 4 times RPM causing band alarms at Position 6V. 4X RPM is the lobe passing freguency. It is common for this frequency to be high from time to time and does not normally represent a problem. All other test point locations are below alarm. Therefore, trend in future surveys.	5

*① = TAKE IMMEDIATE REPAIR ACTION DUE TO SEVERE PROBLEM(S) NOTED (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
2 = SCHEDULE REPAIRS TO CORRECT PROBLEMS NOTED AT CONVENIENCE (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
3 = DESIGNATED PROBLEM DOES NOT WARRANT ACTION AT THIS TIME (CONTINUE ONLY WITH RELIABILITY SURVEY TREND ANALYSIS)

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(704) 333-9011

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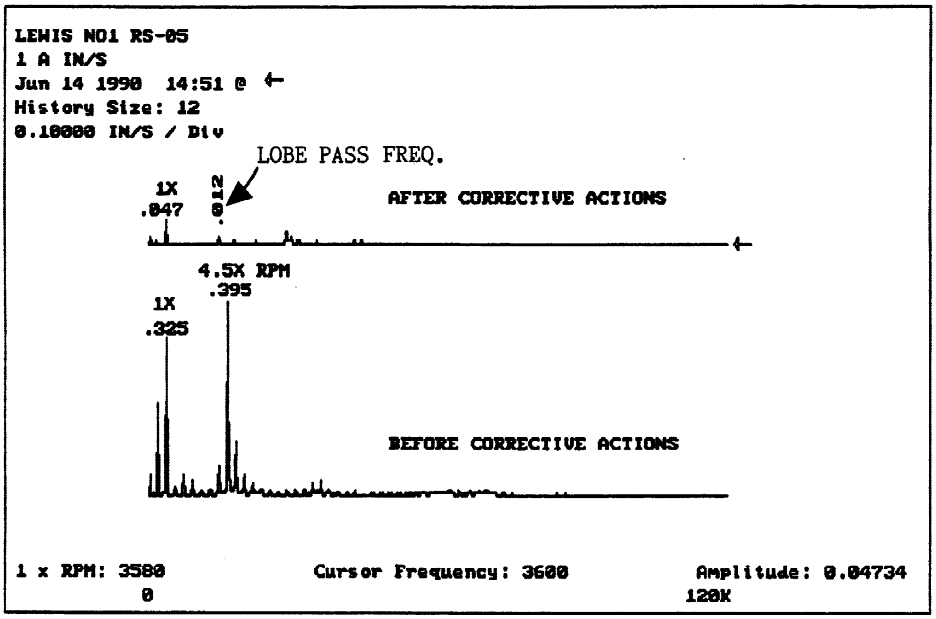
FIGURE 4



001% x 1.0000 = 0.01000 1 x RPM: 3580.00

Machine	Pos	Dir	Units	Date
LEWIS NO1 RS-05	1	A	IN/S	Jun 14 1990 14:51

Frequency	1 x RPM	Amplitude	
300	0.08	0.01493	
3600	1.01	0.04734	
14400	4.02	0.01239	←LOBE PASS
28500	7.96	0.02685	←2X LOBE PASS
29700	8.30	0.01446	

T - Threshold adjust Q - Quit

FIGURE 5
AFTER REPAIR OF SLIDE VALVE

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DETECTION OF SECOND STAGE PLAIN BEARING PROBLEMS ON A CENTRIFUGAL AIR COMPRESSOR

I. BACKGROUND

A fossil fuel plant of a utility contracted Technical Associates to provide and install a complete computer-based predictive maintenance vibration program at each of their plants. In addition, Technical Associates was to collect the initial baseline measurements and issue a baseline report. One of the machines evaluated was the Service Air Centac #61 (SERVAIR CNTC 61) as seen in the machine sketch of Figure 1. Following are results of the evaluation.

II. DISCUSSION OF RESULTS

The latest measurements for this machine are shown in Figure 2 (overall vibration and spike energy levels). Figure 3 is a spectrum that was taken from the machine's 2nd stage non-contact probe permanent monitor. The measurement is taken with a special cable that allows a data collector to connect to the BNC terminals of the monitor panel. Notice in Figure 3 that a Band 4 alarm violation occurred. The peak was at 88,500 CPM which corresponded to 2X the Stage 2 rotational speed of 44,310 RPM. When spectra are captured from non-contact displacement probe monitors on these centrifugal air compressors, 2X RPM amplitudes greater than approximately 30% of the amplitude at 1X RPM become questionable. In this case, the 2X RPM amplitude was over 80% of that at 1X RPM indicating a potential problem (particularly since the amplitude at 4X second stage speed was likewise high.

Since these second stage bearings were sleeve bearings, one of the more likely problems was elevated clearance between the shaft and bearing causing it to be slightly loose, thereby likely allowing the shaft to orbit more than it should within the bearing. This elevated clearance could be due to wear or, in this case, may have been due to the inherent clearance of the bearing. Since this was the initial baseline measurements, no previous spectral data was available for comparison. However, since the amplitudes were relatively low, the condition was not judged to be a serious problem at that time.

III. RECOMMENDATIONS

A. Continue to trend the Stage 2 bearing condition as recommended in the **Baseline Report** section shown in Figure 4.

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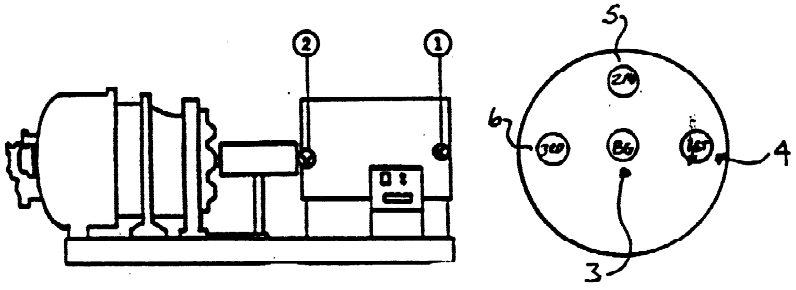
RELIABILITY SURVEY predictive maintenance program (Page__of__)

UNITS
5
6
6T
62

DETECTION OF SECOND STAGE PLAIN BEARING PROBLEMS ON
A CENTRIFUGAL AIR COMPRESSOR



AREA:_____

MACHINE IDENT: SERVICE AIR CENTAC ROUTE:_

DRIVER	INTERMEDIATE	DRIVEN
RPM min : 3565	RPM in : _____	RPM min :_____
max :_____	out : _____	max :_____
MFGR :_____	RATIO:_____	MFGR. :_____
MODEL :_____	MODEL :_____	MODEL :_____
SER NO. :_____	SER NO. :_____	SER NO. :_____
V____ P____ A____	RATING :_____	RATING :_____
SHEAVE :_____DIA.	BELT # :_____	#BLADES :_____
BRG#: A (I)	BRG#:_____(I)	BRG#: 5 (I)
BRG#:_____(O)	BRG#_____(O)	BRG#:_____(O)
H.P.: 400	GEAR T:_____CT.	GEAR T:_____CT.
OTHER:_____	OTHER:_____	OTHER:_____

FIGURE 1

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IRD Mechanalysis Data Analysis System
Dec-04-89

Machine Report (Latest Data) for: SERVAIR CNTC 61

Date	Time	Pos	Dir	Ampl	Units	Change	Alarm	Inspection
Aug-15-89	08:16	1	A	.072	IN/S	-----%	.250	
Aug-15-89	08:16	1	H	.092	IN/S	-----%	.225	
Aug-15-89	08:16	1	H	.042	g/SE	-----%	1.25	
Aug-15-89	08:16	1	O	.077	IN/S	-----%	.250	
Aug-15-89	08:17	1	V	.131	IN/S	-----%	.225	
Aug-15-89	08:17	2	A	.062	IN/S	-----%	.250	
Aug-15-89	08:18	2	H	.071	IN/S	-----%	.225	
Aug-15-89	08:18	2	H	.081	g/SE	-----%	1.25	
Aug-15-89	08:18	2	V	.079	IN/S	-----%	.225	
Aug-15-89	08:19	3	A	.133	IN/S	-----%	.300	
Aug-15-89	08:20	3	H	.068	IN/S	-----%	.300	
Aug-15-89	00:00	3	V	.035	IN/S	-----%	.300	
Aug-15-89	08:23	4	H	.041	IN/S	-----%	.250	
Aug-15-89	08:24	4	V	.043	IN/S	-----%	.250	
Aug-15-89	08:21	5	A	.091	IN/S	-----%	.300	
Aug-15-89	08:21	5	H	.051	IN/S	-----%	.250	
Aug-15-89	08:21	5	V	.054	IN/S	-----%	.250	
Aug-15-89	08:22	6	A	.101	IN/S	-----%	.300	
Aug-15-89	08:22	6	H	.045	IN/S	-----%	.250	
Aug-15-89	08:22	6	V	.038	IN/S	-----%	.250	
Aug-15-89	08:24	3A	A	.149	IN/S	-----%	.300	
Aug-15-89	08:25	3H	H	.062	IN/S	-----%	.300	
Aug-15-89	08:25	3V	V	.045	IN/S	-----%	.300	
Aug-15-89	11:03	M1	H	.236	MILS,L	-----%	.650	
Aug-15-89	11:04	M2	H	.431	MILS,L	-----%	.600	
Aug-15-89	11:04	M3	H	.227	MILS,L	-----%	.600	
Aug-15-89	08:27	OPR	O	9256	Hours	-----%	9999	

27 lines this report

FIGURE 2

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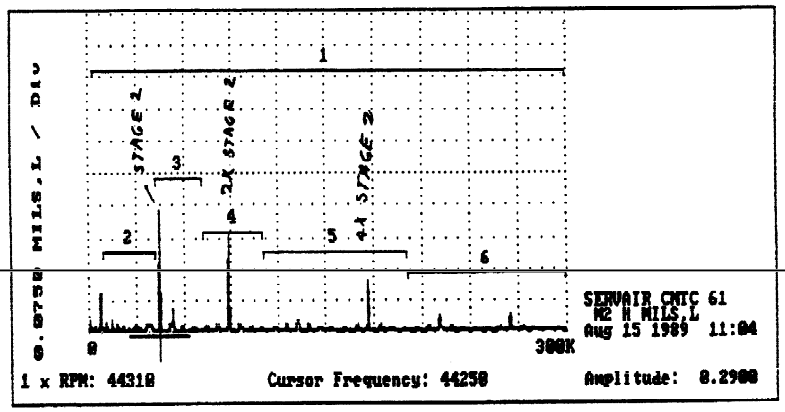

002% x 0.500 = 0.0100 1 x RPM: 44310.00

Machine	Pos	Dir	Units	Date
SERVAIR CNTC 61	M2	H	MILS,L	Aug 15 1989 11:04

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
750	0.017	0.0300	88500	1.997	0.2400
7500	0.169	0.0925	94500	2.133	0.0175
10500	0.237	0.0225	96000	2.167	0.0175
14250	0.322	0.0300	124500	2.810	0.0175
17250	0.389	0.0200	126000	2.844	0.0175
29250	0.660	0.0175	132750	2.996	0.0300
30750	0.694	0.0150	138750	3.131	0.0175
36000	0.812	0.0200	140250	3.165	0.0175
37500	0.846	0.0175	168750	3.808	0.0150
39000	0.880	0.0175	170250	3.842	0.0175
44250	0.999	0.2900	176250	3.978	0.1200
50250	1.134	0.0175	183000	4.130	0.0150
52500	1.185	0.0550	184500	4.164	0.0175
57750	1.303	0.0150	214500	4.841	0.0150
59250	1.337	0.0150	220500	4.976	0.0375
80250	1.811	0.0175	228750	5.162	0.0150
81750	1.845	0.0175	264750	5.975	0.0400

T - Threshold adjust Q - Quit

PLOT 43

FIGURE 3

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SUMMARY OF RESULTS AND RECOMMENDATIONS FOR MACHINES FOUND TO HAVE PROBLEMS

(SORTED FIRST BY PROBLEM PRIORITY; THEN BY PROBLEM SEVERITY WITHIN PRIORITY)

AS OF: 8/24/89

AREA	MACHINE COMPUTER NAME	INDICATED PROBLEM(S): UNBALANCE	MISALIGNMENT/SOFT FOOT	BENT SHAFT	LUBRICATION PROBLEM	BEARING PROBLEM	MECHAN. LOOSENESS	BELT/SHEAVE PROBLEM	RESONANCE PROBLEM	GEAR PROBLEM	BLADE PASSING PROB.	CAVITATION	MOTOR ELECTRICAL	OIL WHIRL /WHIP	LOOSE FOUNDATION	VIB. MONITOR PROB.	SEAL /ROTOR RUB	OTHER (SEE REMARKS)	*PRIORITY OF PROBLEM	HIGHEST OVERALL VIBRATION: POSITION NO.	HIGHEST OVERALL VIBRATION: LEVEL (m = mils, i = in/sec, g = gs)	RECOMMENDATIONS & REMARKS	SEE PLOT NO.
	AC TURN VPMP U6												●						3	1N-0	.094i	Probable rotor bar passing frequency (103,500 CPM) noted at the motor indicating incipient motor electrical problem. Trend in future surveys.	2
	SILO VENT FAN62					●													3	2V	.093i	Band 6 alarm at Position 2V on the motor indicating incipient bearing wear. Trend in future surveys.	53
	I.D. FAN NO. 61										●		●						3	1H	.090i	Band alarm at 7200 CPM (two times line frequency) at Position 1H on the motor. Likely due to a motor electrical problem. Could also be blade passing frequency of the fan if it has 10 blades. Trend in future surveys.	31
	FW HTRDRNS P61A												●						3	1-0	.088i	Band 2 alarm due to possible rotor bar passing frequency as seen on Plot 24. If the motor has anti-friction bearings it is likely due to incipient bearing wear. Trend in future surveys.	24
	SERVAIR CNTC 61					●	●												3	2-H 5H	.255 MILS,L .051i	Band 4 alarm on stage two of the compressor due to harmonics of second stage running speed. Possible mechanical looseness due to bearing wear on the 2nd stage. Trend in future surveys.	43

*① = TAKE IMMEDIATE REPAIR ACTION DUE TO SEVERE PROBLEM(S) NOTED. (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
2 = SCHEDULE REPAIRS TO CORRECT PROBLEMS NOTED AT CONVENIENCE. (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
3 = DESIGNATED PROBLEM DOES NOT WARRANT ACTION AT THIS TIME. (CONTINUE ONLY WITH RELIABILITY SURVEY TREND ANALYSIS)

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FIGURE 4

DETECTION OF PLAIN BEARING PROBLEMS IN THE MOTOR SERVING A CENTRIFUGAL AIR COMPRESSOR

I. BACKGROUND

Technical Associates was selected by a utility company to fully set up and implement a predictive maintenance vibration program at each of its fossil fuel plants. Part of the contract included the collection of baseline measurements and providing a baseline condition evaluation report. One of the machines included in the baseline was a three stage Ingersoll-Rand Centac centrifugal air compressor identified by the plant as Service Air Centac 62. The condition evaluation of this machine is summarized below.

II. DISCUSSION OF RESULTS

A sketch for this machine is shown in Figure 1 and a Latest Measurements Machine Report in Figure 2. Both spectral band and overall alarms were triggered at each motor bearing. In addition, a spectral band alarm was exceeded at the compressor's bullgear bearing (Position 3). Figures 3 and 4 show that the outboard motor bearing (Position 1) alarms were due to the presence of many harmonics of motor speed which is typical of either mechanical looseness or the latter stages of sleeve bearing wear when clearances between the bearing and journal increase. Note the distinct difference in the spectrum taken on the inboard motor bearing shown in Figure 5. Unlike Figures 3 and 4, Figure 5 showed very little running speed harmonic activity. Instead, Figure 5 showed a high amplitude peak at 1X RPM of .290 in/sec, along with a still noticeable .095 in/sec peak at 2X RPM. What remained were very low amplitude harmonics of RPM. The Position 3 bullgear spectrum of Figure 6 also had several small bullgear speed harmonics, along with peaks at the first, second and third stage running speeds present which caused the RSS energy within Band 4 to exceed the band alarm.

III. CONCLUSIONS

The multiple harmonics seen on the motor of SERVAIR CNTC 62 were typical of mechanical looseness. Since the motor had plain bearings, the many harmonics seen in Figures 3 and 4 can result from bearing wear causing a mechanical looseness condition due to higher than normal clearances between the motor shaft and bearing journal. The imbalance (1X RPM amplitude of .290 in/sec) seen in Figure 5 may have resulted from misalignment, an exaggerated mechanical looseness condition which was amplifying a small unbalance and/or a couple unbalance with more unbalance at the inboard motor bearing than at the outboard motor bearing.

IV. RECOMMENDATIONS

As seen in Figure 7 (a page from the **Baseline Report**), Technical Associates recommended this machine be watched closely for any future changes that would signal additional deterioration of the motor bearing(s).

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RELIABILITY SURVEY predictive maintenance program (Page__of__)

UNITS
5
6
61
(62)

DETECTION OF PLAIN BEARING PROBLEMS IN THE MOTOR SERVING A CENTRIFUGAL AIR COMPRESSOR

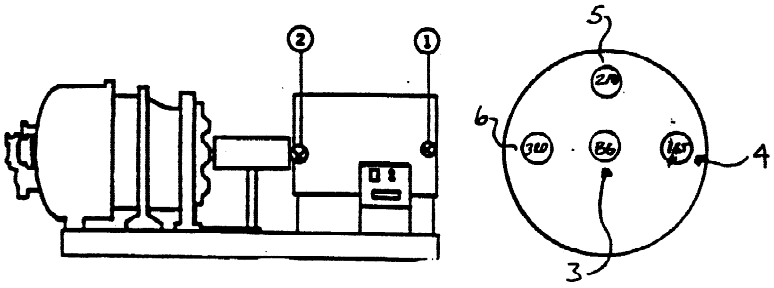

FIGURE 1.

MACHINE IDENT: SERVICE AIR CENTAC

AREA:

ROUTE:

DRIVER	INTERMEDIATE	DRIVEN
RPM min : 3565	RPM in :	RPM min :
max :	out :	max :
MFGR :	RATIO:	MFGR. :
MODEL :	MODEL :	MODEL :
SER NO. :	SER NO. :	SER NO. :
V___ P___ A___	RATING :	RATING :
SHEAVE : DIA.	BELT # :	#BLADES :
BRG#: A (I)	BRG#: (I)	BRG#: 5 (I)
BRG#: (O)	BRG# (O)	BRG#: (O)
H.P.: 400	GEAR T: CT.	GEAR T: CT.
OTHER:	OTHER:	OTHER:

FIGURE 1

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IRD Mechanalysis Data Analysis System Page 1
Dec-04-89

Machine Report (Latest Data) for: SERVAIR CNTC 62

Date	Time	Pos	Dir	Ampl	Units	Change	Alarm	Inspection
Aug-24-89	15:14	1	A	.172	IN/S	-----%	.250	
Aug-24-89	15:14	1	H	.179	IN/S	-----%	.225	
Aug-24-89	15:14	1	H	.145	g/SE	-----%	1.25	
Aug-24-89	15:15	1	O	.206	IN/S	-----%	.250	
Aug-24-89	15:15	1	V	*.246	IN/S	-----%	.225	
Aug-24-89	15:15	2	A	.135	IN/S	-----%	.250	
Aug-24-89	15:16	2	H	*.327	IN/S	-----%	.225	
Aug-24-89	15:16	2	H	.144	g/SE	-----%	1.25	
Aug-24-89	15:16	2	V	.167	IN/S	-----%	.225	
Aug-24-89	15:16	3	A	.223	IN/S	-----%	.300	
Aug-24-89	15:16	3	H	.065	IN/S	-----%	.300	
Aug-24-89	15:16	3	V	.047	IN/S	-----%	.300	
Aug-24-89	15:20	4	A	.082	IN/S	-----%	.300	
Aug-24-89	15:20	4	H	.050	IN/S	-----%	.250	
Aug-24-89	15:20	4	V	.047	IN/S	-----%	.250	
Aug-24-89	15:20	5	A	.090	IN/S	-----%	.300	
Aug-24-89	15:20	5	H	.077	IN/S	-----%	.250	
Aug-24-89	15:20	5	V	.108	IN/S	-----%	.250	
Aug-24-89	15:19	6	A	.076	IN/S	-----%	.300	
Aug-24-89	15:19	6	H	.072	IN/S	-----%	.250	
Aug-24-89	15:19	6	V	.062	IN/S	-----%	.250	
Aug-24-89	15:18	3A	A	.289	IN/S	-----%	.300	
Aug-24-89	15:18	3H	H	.082	IN/S	-----%	.300	
Aug-24-89	15:18	3V	V	.049	IN/S	-----%	.300	
Aug-24-89	15:23	M1	H	.314	MILS,L	-----%	.650	
Aug-24-89	15:23	M2	H	.298	MILS,L	-----%	.600	
Aug-24-89	15:23	M3	H	.187	MILS,L	-----%	.600	
Aug-24-89	15:26	OPR	O	1145	Hours	-----%	9999	

28 lines this report

FIGURE 2

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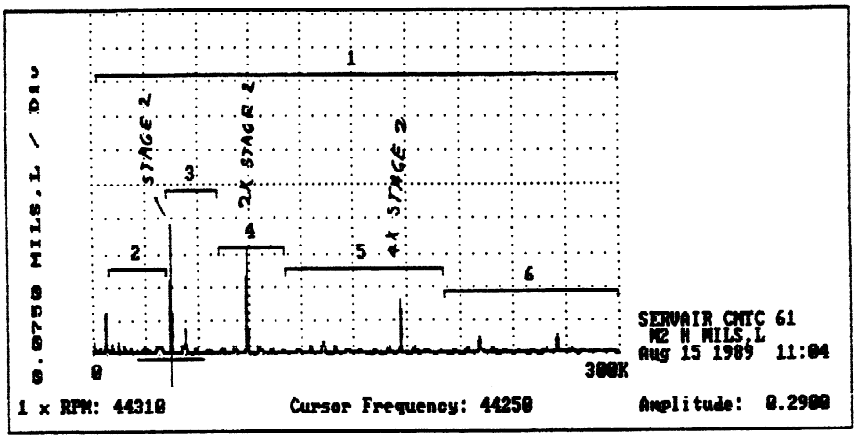

002% x 0.500 = 0.0100 1 x RPM: 44310.00

Machine	Pos	Dir	Units	Date
SERVAIR CNTC 61	M2	H	MILS,L	Aug 15 1989 11:04

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
750	0.017	0.0300	88500	1.997	0.2400
7500	0.169	0.0925	94500	2.133	0.0175
10500	0.237	0.0225	96000	2.167	0.0175
14250	0.322	0.0300	124500	2.810	0.0175
17250	0.389	0.0200	126000	2.844	0.0175
29250	0.660	0.0175	132750	2.996	0.0300
30750	0.694	0.0150	138750	3.131	0.0175
36000	0.812	0.0200	140250	3.165	0.0175
37500	0.846	0.0175	168750	3.808	0.0150
39000	0.880	0.0175	170250	3.842	0.0175
44250	0.999	0.2900	176250	3.978	0.1200
50250	1.134	0.0175	183000	4.130	0.0150
52500	1.185	0.0550	184500	4.164	0.0175
57750	1.303	0.0150	214500	4.841	0.0150
59250	1.337	0.0150	220500	4.976	0.0375
80250	1.811	0.0175	228750	5.162	0.0150
81750	1.845	0.0175	264750	5.975	0.0400

T - Threshold adjust Q - Quit

PLOT 44

FIGURE 3

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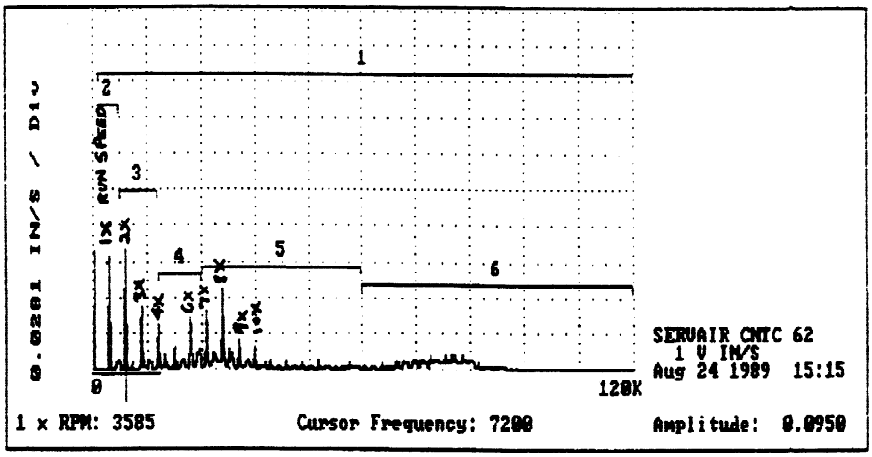

002% x 0.500 = 0.0100 1 x RPM: 3585.00

Machine	Pos	Dir	Units	Date
SERVAIR CNTC 62	1	V	IN/S	Aug 24 1989 15:15

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
300	0.08	0.0925	80100	22.34	0.0100
3600	1.00	0.0900	80700	22.51	0.0100
7200	2.01	0.0950	82500	23.01	0.0100
10800	3.01	0.0500			
14400	4.02	0.0350			
15900	4.44	0.0100			
18000	5.02	0.0175			
21300	5.94	0.0400			
23400	6.53	0.0150			
24900	6.95	0.0450			
26700	7.45	0.0125			
28500	7.95	0.0625			
30000	8.37	0.0150			
32100	8.95	0.0225			
35700	9.96	0.0175			
75300	21.00	0.0100			
78600	21.92	0.0100			

T - Threshold adjust Q - Quit

PLOT 45

FIGURE 4

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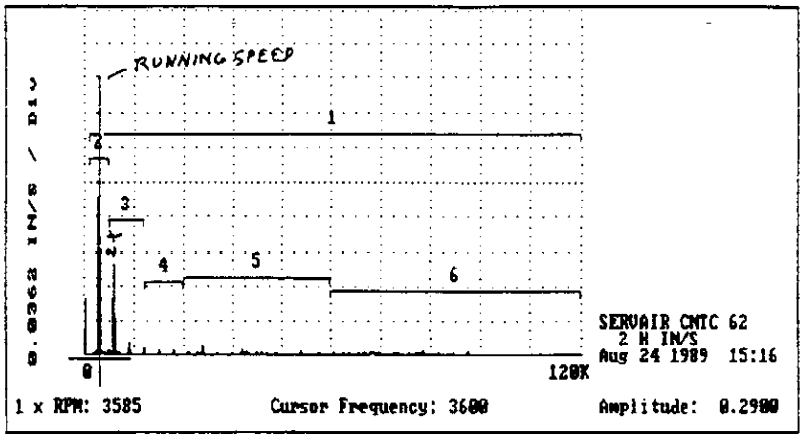

002% x 0.500 = 0.0100 1 x RPM: 3585.00

Machine	Pos	Dir	Units	Date
SERVAIR CNTC 62	2	H	IN/S	Aug 24 1989 15:16

Frequency	1 x RPM	Amplitude
300	0.08	0.0600
3600	1.00	0.2900
7200	2.01	0.0950
10800	3.01	0.0125
28500	7.95	0.0100

T - Threshold adjust Q - Quit

PLOT 46

FIGURE 5

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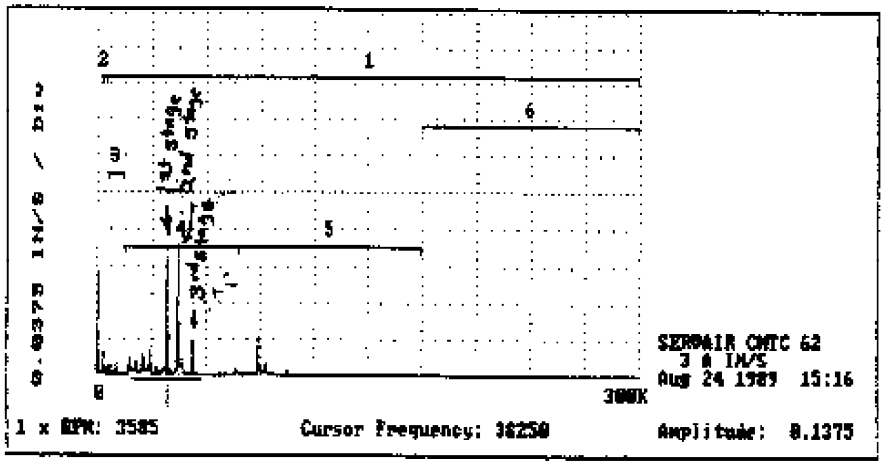

002% = 0.500 = 0.0100 1 x RPM: 3585.00

Machine	Pos	Dir	Units	Date
SERVAIR CNTC 62	3	A	IN/S	Aug 24 1989 15:16

Frequency	X = RPM	Amplitude
750	0.21	0.1075
3750	1.05	0.0225
7500	2.09	0.0100
10500	2.93	0.0125
18000	5.02	0.0175
21750	6.07	0.0150
24750	6.90	0.0200
28500	7.95	0.0250
32250	9.00	0.0100
38250	10.67	0.1375
44250	12.34	0.1350
46500	12.97	0.0175
52500	14.64	0.0350
88500	24.69	0.0400
93000	25.94	0.0125

T = Threshold adjust Q = Quit

PLOT 47

FIGURE 6

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SUMMARY OF RESULTS AND RECOMMENDATIONS FOR MACHINES FOUND TO HAVE PROBLEMS

(SORTED FIRST BY PROBLEM PRIORITY; THEN BY PROBLEM SEVERITY WITHIN PRIORITY)

AS OF: 8/24/89

AREA	MACHINE COMPUTER NAME	INDICATED PROBLEM(S)																			*PRIORITY OF PROBLEM	HIGHEST OVERALL VIBRATION		RECOMMENDATIONS & REMARKS	SEE PLOT NO.
		UNBALANCE	MISALIGNMENT/SOFT FOOT	BENT SHAFT	LUBRICATION PROBLEM	BEARING PROBLEM	MECHAN. LOOSENESS	BELTS/SHEAVE PROBLEM	RESONANCE PROBLEM	GEAR PROBLEM	BLADE PASSING PROB.	CAVITATION	MOTOR ELECTRICAL	OIL WHIRL/WHIP	LOOSE FOUNDATION	VIB. MONITOR PROB.	SEAL /MOTOR RUB	OTHER (SEE REMARK(S))		POSITION NO.	LEVEL (m = mils, i = in/sec, g = g's)			
	CONST DIF PMP52					●	●													2	2V	.144i	Band 5 alarm at Position 2 on the motor due to mechanical looseness. Inspect for loosened fasteners, broken structures and bearing wear or looseness. Make appropriate repairs. Replace bearing if bearing related. Trend in future surveys.	10
	CONST DIF PMP62					●	●													2	2H	.131i	Band 4 alarm at Position 2H due to mechanical looseness as seen on Plot 11. Inspect for loosened fasteners, broken structures and bearing wear or looseness. Make appropriate repairs. Replace bearing if bearing related. Trend in future surveys.	11
	INST AIR RECP61									●										3	1A 3H 3H	.400i .732 g-SE .115i	Band alarm at Position 1A due to resonant amplification of five (5) times the compressor RPM (5 X 420 CPM). See Plot 33. Spike energy alarm at Position 3H on the compressor, suggests lubrication should be checked. Trend in future surveys.	33
	I.D. FAN NO. 51		●																	3	2A	.376i	Band 4 alarm due to misalignment. Suggest precision aligning the unit.	29
	FW HTRDRN3 P61A	●	●																	3	3H	.347i	Large one (1) times running speed vibration likely due to unbalance or misalignment. Some mechanical looseness also noted. Trend in future surveys.	23
	SERVAIR CNTC 62	●	●			●	●													3	1A 1V 2H 3A	.172i .246i .327i .223i	Many harmonics of motor/bullgear RPM throughout Positions 1, 2 and 3. Likely the result of some unbalance, misalignment and mechanical looseness. Also possible wear of motor bearings indicated by large number of harmonics in Plots 44 and 45. Watch closely. Trend in future surveys.	44 45 46 47

*① = TAKE IMMEDIATE REPAIR ACTION DUE TO SEVERE PROBLEM(S) NOTED. (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
2 = SCHEDULE REPAIRS TO CORRECT PROBLEMS NOTED AT CONVENIENCE. (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
3 = DESIGNATED PROBLEM DOES NOT WARRANT ACTION AT THIS TIME. (CONTINUE ONLY WITH RELIABILITY SURVEY TREND ANALYSIS)

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FIGURE 7

DETECTION OF SERIOUS WEAR OF A CENTRIFUGAL COMPRESSOR BULLGEAR SKF 6213 BALL BEARING

I. BACKGROUND

An annual contract was awarded Technical Associates beginning January, 1984 encompassing 52 machines including several Ingersoll-Rand Centac centrifugal air compressors. The primary instrument used at that time was a Hewlett Packard model 3561A real-time analyzer. Measurements were taken at 4-week intervals on each of the machines. Figures 1 thru 3 include drawings showing one of the Ingersoll-Rand Centac compressors (in this case, a model C-21 having an output capacity of 2100 ICFM). Figure 1 shows that measurements were taken on each of the motor bearing housings as well as on the bullgear (Position 3) and each of the four impeller rotor bearing housings (Positions 4 thru 7). Note the pictorial view of the bullgear and each of its mating pinions on Figure 1. The bullgear operated at about 3580 RPM (59.67 Hz), whereas the impeller rotor speeds ranged from 29,325 RPM (488.75 Hz) for the 1st Stage up to 49,270 RPM (821.13 Hz) in the case of the 4th Stage.

The problem with these machines is that once a defect occurs on either a bullgear or on one of its mating pinions, it can propagate rapidly from one component to another. Note that the cost for rebuilt rotors at that time (1984) ranged from about $8,000 for the 4th Stage up to around $26,000 for the 1st Stage. Thus, it was most important to detect problems in initial stages and take corrective actions before the problems were allowed to transmit from one component to another.

Figure 2 shows a typical cross section of one of the impeller rotors, each of which were outfitted with proximity probes which were serving permanent vibration monitors. The Alarm level for these machines was set at .70 mil (Pk-Pk) whereas the Trip level was at 1.0 mil (as per the manufacturer).

It is important to point out that each of three types of measurements were taken on these machines including:

A. **Vibration Displacement Measurements Directly from Proximity Probe Monitors -** a cable is connected directly to the outputs of each of the four proximity probes serving each of the four impeller stages. This cable is then connected directly to the HP 3561A analyzer allowing the capture not only of overall displacement (which is displayed by the monitor), but also enables the generation of complete vibration spectra of the shaft vibration at each of these stages.

B. **Vibration Velocity Measurements from each Bearing Housing** - a standard accelerometer having a frequency range of 1.0 to 10,000 Hz (60 - 600,000 CPM) is mounted on each of the bearing housings in the locations shown in Figure 1. A number of different frequency range spectra are captured on the various bearing housings, depending on the speed of the shaft at that location. The acceleration signal from the accelerometer is integrated to velocity within the HP 3561A analyzer (from peak acceleration to peak velocity).

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C. **High Frequency Acceleration Measurement from Compressor Bearing Housings** - a special, high frequency accelerometer is employed along with signal conditioning instrumentation in order to capture measurements up to at least 2X gear mesh frequency (2 GMF). Table I shows that the fundamental gear mesh frequency (GMF) is 20,528 Hz (1,231,700 CPM). Because measurements need to be taken up to 2 and 3X gear mesh frequency (2 GMF and 3 GMF), this puts great demands on the transducer system as well as the mounting of the accelerometer onto the housing. Great attention to each of these items has enabled measurements with repeatable amplitudes up thru 3 GMF (61,585 Hz) although the absolute value of the amplitude itself at this high frequency might be in question. That is, if the accelerometer is mounted and measurements taken several different times, it is possible to get good repeatable acceleration measurements. However, if the system displays 10 g acceleration, the true level might be 7 or 8 g, or on the other hand, it might be 12 to 14 g. In any case, due to the good repeatability of the measurement, data taken from one month to the next is successfully trended. In fact, these same procedures were used on approximately 30 different Centac compressors on a monthly basis during 1984 by Technical Associates and have been used on several hundred additional Centacs since that time.

II. ANALYSIS RESULTS

Figure 3 includes a section view showing where measurements are taken in the axial direction on the bullgear (Position 3A), 2nd Stage (Position 5A) and 3rd Stage (Position 6A). Note the location of each of the bullgear ball bearings in Figure 3. The SKF 6213 bearing is inboard (closest to the coupling) whereas the SKF 7213 is somewhat remote from the measurement location. Note that the vibration signal from the SKF 6213 bearing transmits cleanly from the housing through the casing into the casing bolt on which the accelerometer is mounted. On the other hand, the vibration from the SKF 7213 bearing must travel along the shaft and then through either the SKF 6213 bearing and/or machine casing. Therefore, a number of experiments over the years on these machines have shown suppressed amplitudes which must be taken into account that come from bearing defect frequencies from the SKF 7213 bearing relative to those of the SKF 6213.

Table I provides many of the anticipated vibration frequencies generated by components of the compressor. These include the gear mesh frequency (GMF), the blade pass frequencies of each of the 4 Stages (BPF), each of the impeller rotor speeds (F_1 thru F_4), the main oil pump frequency (F_{PUMP}), the SKF 7213 bearing defect frequencies, the SKF 6213 bearing defect frequencies, and finally the bullgear and motor running speed frequency (F_M). Note that this table includes the first 10 harmonics of each of these frequencies (1F thru 10F).

Figure 4 shows one of these spectra captured approximately 9 months after initial baselines were captured. Note that several frequencies appear on this spectrum including the 2nd Stage running speed peak (F_2), the 3rd Stage speed (F_3) and the 4th Stage speed (F_4). Also showing up was a high amplitude peak at approximately 352 Hz. Referring back to Table I, note that the 352 Hz frequency corresponds to the inner race defect frequency of the SKF 6213 bearing (6213 BPFI).

Importantly, when initial baseline signatures were captured in January, 1984, this 352 Hz peak appeared with an amplitude of .216 in/sec and did not have bullgear running speed sidebands to its left and right (which later did appear as shown in Figure 4). Due to the baseline amplitude of over .20 in/sec which is quite high for a bearing defect frequency, Technical Associates recommended replacement of the bullgear bearing immediately before the problem, then isolated only with the bullgear bearing, caused problems with the mating components. Due to great demands on this compressor and its support of critical process machinery, the plant decided it could not shut down the machine to accomplish the bearing replacement at this time. Therefore, the decision was made to simply trend future spectra on this machine and try to schedule this replacement along with the production department

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ASAP. Three months later, the amplitude of this frequency began climbing and went up to about .252 in/sec as of April, 1984. At this point, small amplitude sideband frequencies at bullgear speed began to appear to the left and right of this bearing frequency. Once again, Technical Associates recommended bullgear bearing replacement even though the other measurements on this machine did not indicate any damage to other machine components yet (for example, acceleration measurements still showed that no gear mesh frequencies exceeded about 7 g which is normal for these machines at these GMF frequencies and harmonics). Still the plant felt it could not schedule the bearing replacement.

By June, amplitudes at BPFI (352 Hz) increased from .252 in/sec in April up to .305 in/sec and sidebands at 292 and 410 Hz were now well developed with amplitudes of almost .05 in/sec each. Because amplitudes at gear mesh frequencies had now begun to increase from about 7 g in both January and April up to approximately 12 g now in June, Technical Associates recommended immediate bullgear bearing replacement before significant damage was done to the impeller rotors. However, once again, the plant production department would not allow the machine to be brought down for maintenance.

Therefore, signatures later captured on September 28 showed extensive problems throughout the compressor. Figure 4 shows that the BPFI amplitude was now up to .402 in/sec and had well developed running speed sidebands at 292.5 Hz and 410.0 Hz. In addition, it now showed a small second harmonic inner race frequency at 705 Hz. Of even greater concern were the acceleration spectra captured on the bullgear and pinion housings which are shown in Figures 6 and 7. Figure 6 was a wideband plot from 0 to 100,000 Hz (6,000,000 CPM) and showed that levels at twice gear mesh frequency (2 GMF) had now climbed from only 7 g at baseline up to 42.1 g. Then, looking at the zoom spectrum shown in Figure 7, note that 2 GMF was now surrounded by well developed sidebands at 1st Stage speed indicating significant wear of the 1st Stage pinion. Subsequent measurements on the other stages indicated problems with at least three of the pinions as well as the bullgear itself.

III. RECOMMENDATIONS MADE

As of the September 28 data, Technical Associates recommended replacement of each of the bullgear bearings. Also, it was recommended that each of the gears be closely inspected for suspected wear.

IV. CORRECTIVE ACTIONS TAKEN

When inspection of the gears was made, significant wear was found on the bullgear as well as the 1st, 3rd and 4th Stage pinions. As a result, each of these impeller rotor pinion assemblies had to be replaced as well as the bullgear and its bearings. Figures 5 and 8 were captured after corrective actions were taken. Figure 5 shows that the velocity spectrum showed a dramatic drop in inner race frequency from .402 in/sec down to .015 in/sec after replacement of the SKF 6213 bearing. Figure 8 shows that acceleration levels at 2 GMF dropped from 42.1 g before to 7.62 g after replacement of 4 of the 5 gears. One of the hard lessons learned from this case history was that corrective actions should be taken on these centrifugal compressors when vibration spectra indicate if this is at all possible. What would have been less than a $10,000 repair cost had actions been taken right after the January baseline ended up being repair costs of slightly over $100,000 since the problem was allowed to propagate to so many other components. Fortunately, after this case history, the plant has paid close attention to vibration spectra and successfully detected each of 3 potentially serious incidents that could have resulted in similar costly failures. However, on each of these occasions, the maintenance department coordinated with the production department and took actions which prevented any of the failure scenario costs from exceeding $20,000.

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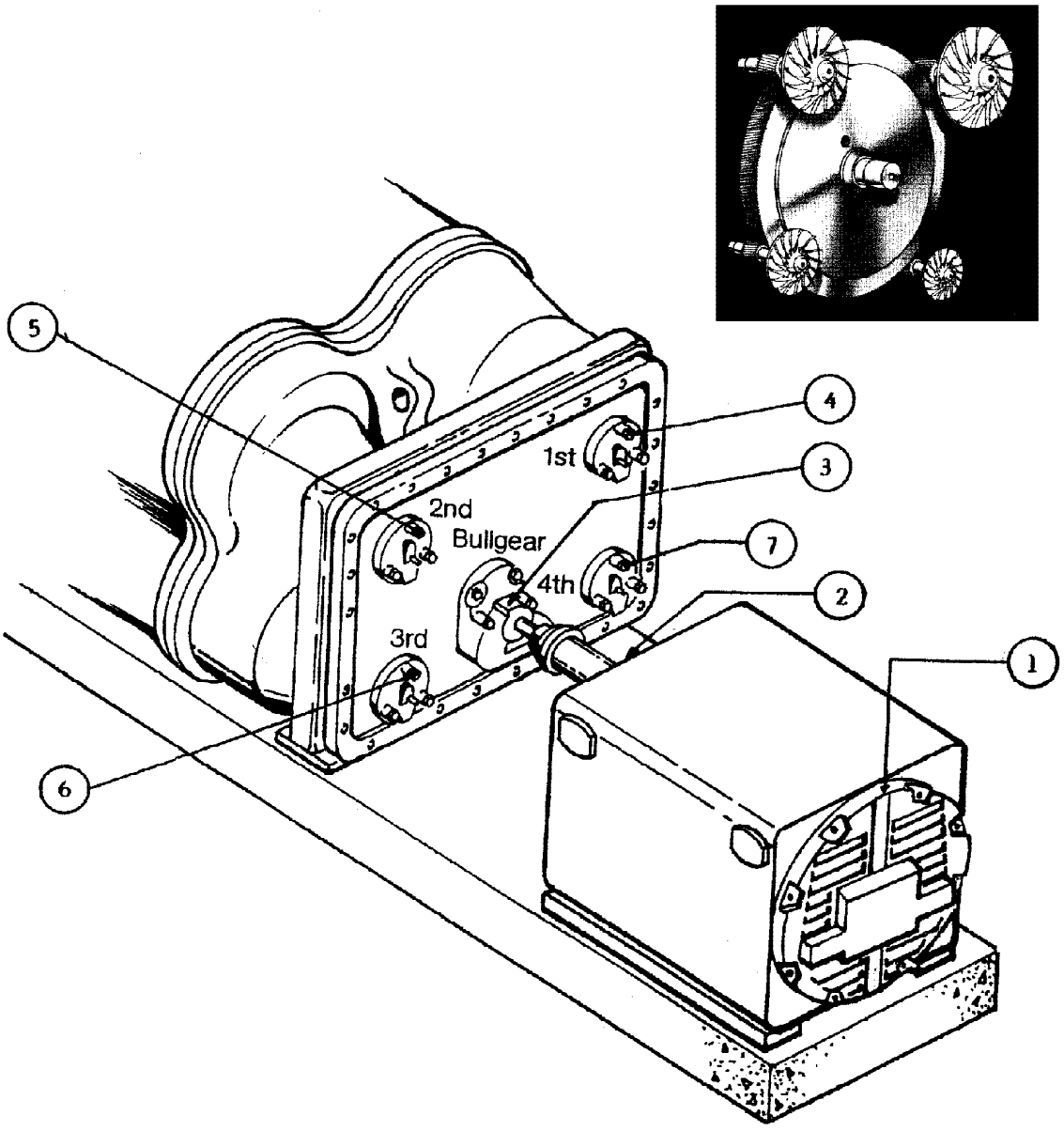

FIGURE 1: ACCELEROMETER LOCATIONS ON AN INGERSOLL-RAND CENTAC CENTRIFUGAL AIR COMPRESSOR (MODEL C-21 2100 ICFM)

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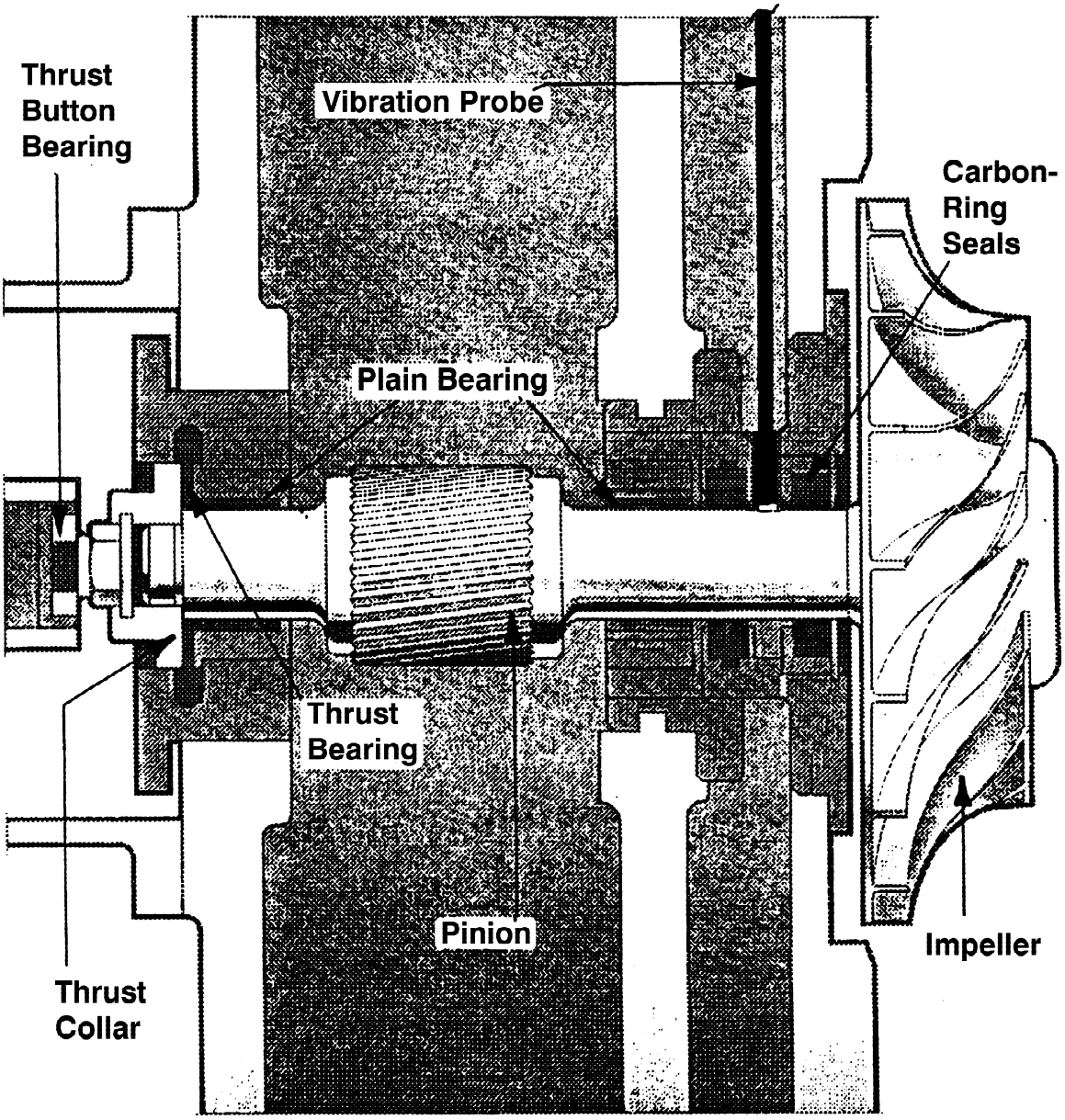
FIGURE 2: INGERSOLL-RAND CENTAC IMPELLER ROTOR CROSS SECTION



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DETECTION OF SERIOUS WEAR OF A CENTAC BULLGEAR INBOARD BALL BEARING

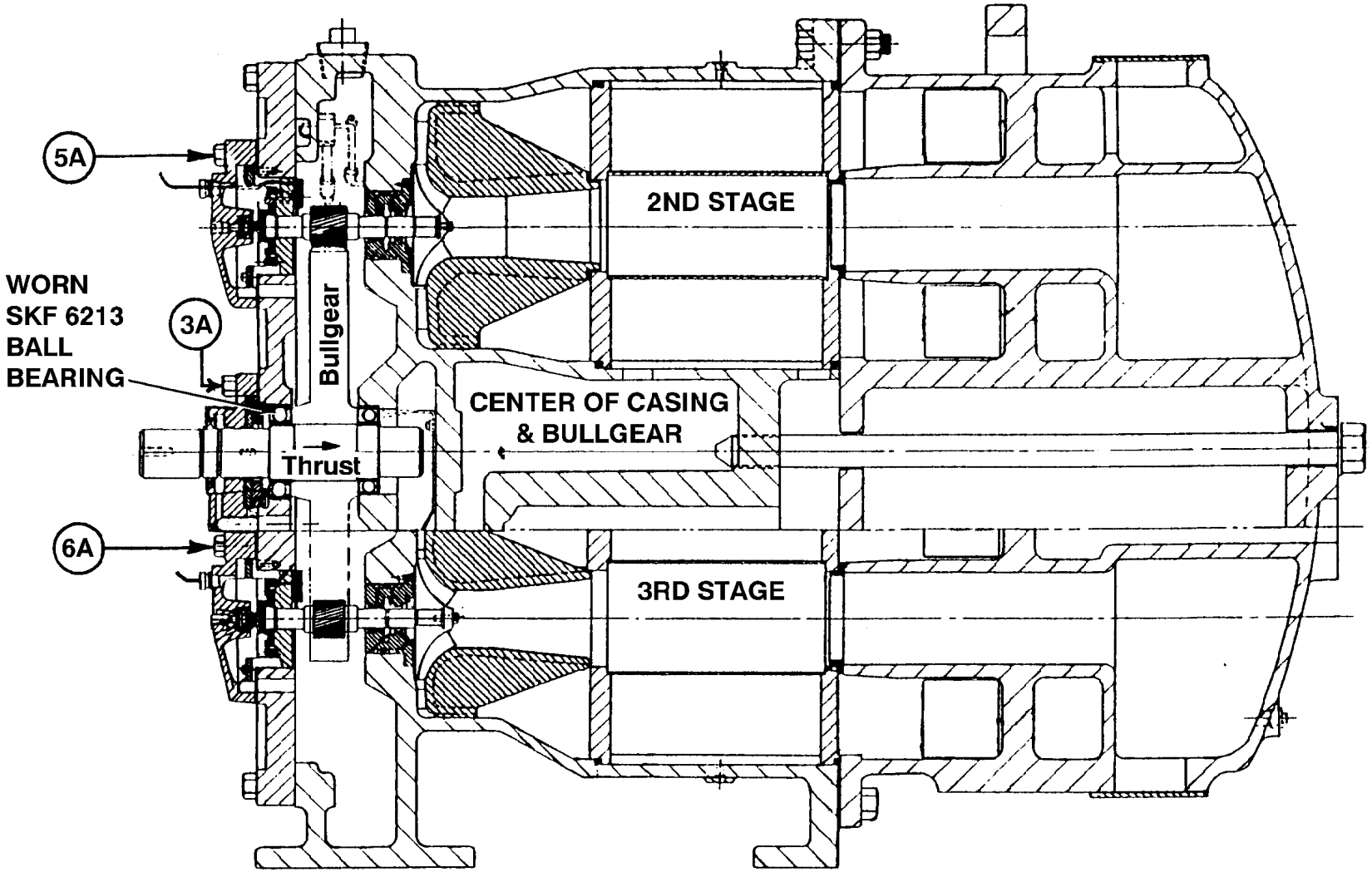

FIGURE 3. SECTION VIEW OF AN INGERSOLL-RAND CENTAC CENTRIFUGAL AIR COMPRESSOR (FRAME 1)

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TABLE 1. ANTICIPATED FREQUENCIES FOR MACHINE #6 C-21 CENTAC (HERTZ)

ITEM FREQ.	$\frac{F_i}{F_M}$	1F	2F	3F	4F	5F	6F	7F	8F	9F	10F
GMF.....	$344F_M$	20,528	41,056	61,585	82,113	102,641	N/A	N/A	N/A	N/A	N/A
BPF_4.....	178.88F	10,675	21,349	32,024	42,699	53,373	64,048	74,723	85,397	96,072	106,747
BPF_3.....	165.63F	9,884	19,768	29,652	39,536	49,420	59,304	69,188	79,072	88,956	98,839
BPF_2.....	150.50F	8,981	17,962	26,943	35,924	44,965	53,886	62,868	71,849	80,830	89,811
BPF_1.....	122.86F	7,331	14,663	21,995	29,326	36,658	43,989	51,320	58,652	65,983	73,315
F_4.....	13.760F	821.13	1642.3	2463.4	3284.5	4105.6	4926.8	5747.9	6569.0	7390.2	8211.3
F_3.....	12.741F	760.30	1520.6	2280.9	3041.2	3801.5	4561.8	5322.1	6082.4	6842.7	7603.0
F_2.....	10.750F	641.51	1283.0	1924.5	2566.0	3207.5	3849.0	4490.5	5132.1	5773.6	6415.1
F_1.....	8.190F	488.8	977.5	1466.3	1955.1	2443.8	2932.6	3421.4	3910.1	4398.9	4887.7
F_{pump}	$13F_M$	775.78	1551.6	2327.3	3103.1	3878.9	4654.7	5430.4	6206.2	6982.0	7757.8
7213 BPFI	9.18F	547.81	1095.6	1643.4	2191.2	2739.0	3286.8	3834.6	4382.4	4930.2	5478.0
7213 BPFO	6.84F	408.18	816.36	1224.54	1632.72	2040.9	2449.08	2857.26	3265.44	3673.62	4081.8
7213 BSF	2.71F	161.72	322.4	483.6	644.8	806.0	967.2	1128.4	1289.6	1450.8	1612.0
7213 FTF	0.426F	25.42	50.84	76.26	101.68	127.1	152.52	177.94	203.36	228.78	254.2
6213 BPFI	5.898F	351.96	703.92	1055.88	1407.84	1759.8	2111.76	2463.72	2815.68	3167.64	3519.6
6213 BPFO	4.098F	244.55	489.1	733.65	978.2	1222.75	1467.3	1711.85	1956.4	2200.95	2445.5
6213 BSF	2.680F	159.93	319.86	479.79	639.72	799.65	959.58	1119.51	1279.44	1439.37	1599.3
6213 FTF	0.410F	24.47	48.94	73.41	97.88	122.35	146.82	171.29	195.76	220.23	244.7
* F_M.....	$1F_M$	59.675	119.35	179.03	238.70	298.38	358.05	417.73	477.40	537.08	596.75

* These calculations are based on a motor running speed of 3580 rpm = 59.675 Hz.

NORMAL OPERATING SPEEDS

COMPONENT	SYMBOL	RPM	Hz
4th Stage	F_4	49,270	821.13
3rd Stage	F_3	45,620	760.30
2nd Stage	F_2	38,490	641.51
1st Stage	F_1	29,325	488.75
Motor	F_M	3,580	59.68

* NOTE: See "Terminology" on page 22 for definitions of "ITEM FREQUENCIES" in above Table VIII.

FIGURE 4: SEVERE DEFECT CONCENTRATED ON ONE LOCATION OF THE INNER RACE OF THE SKF 6213 BULLGEAR BEARING (9/28/84)

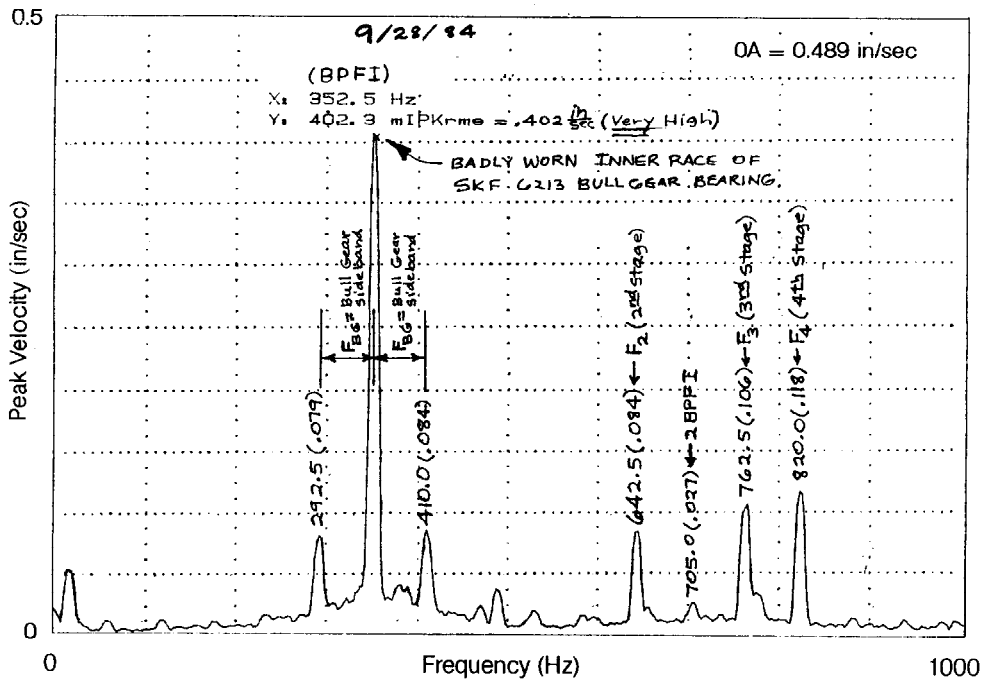
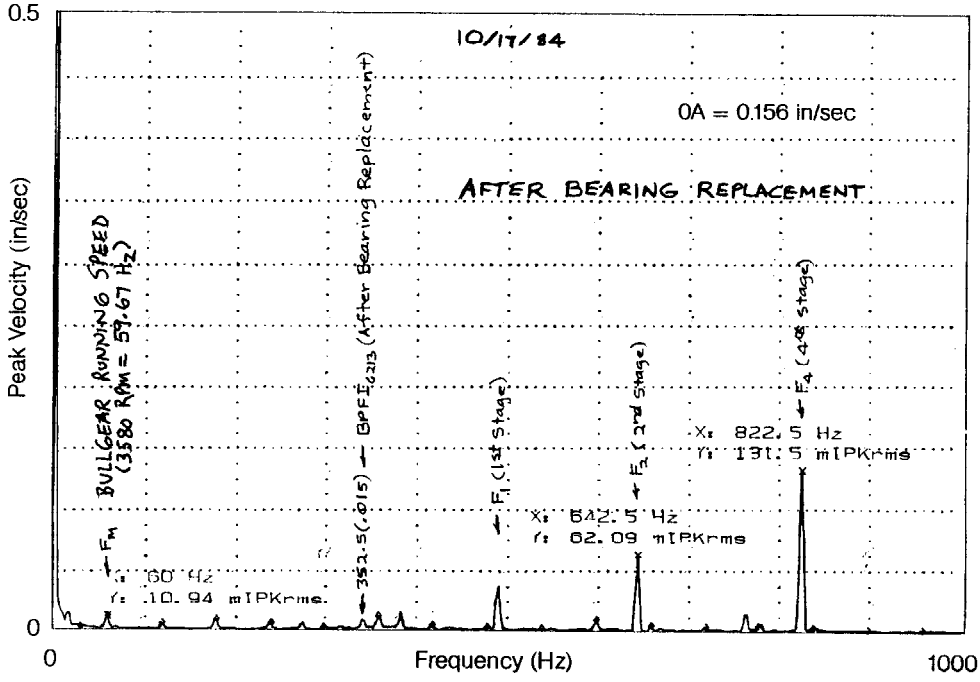


FIGURE 5: AFTER REPLACEMENT OF SKF 6213 BULLGEAR BEARINGS (10/16/84)



INGERSOLL-RAND CENTAC BULLGEAR BALL BEARING CAUGHT JUST PRIOR TO FAILURE

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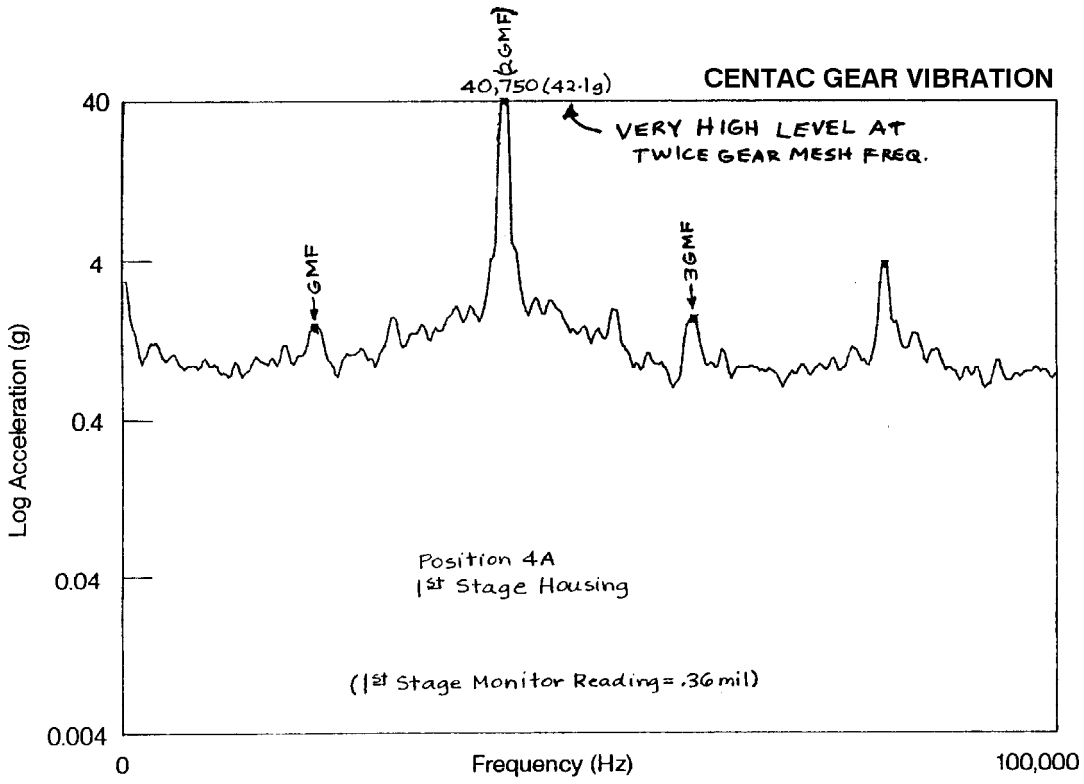

FIGURE 6: HIGH ACCELERATION DUE TO GEAR WEAR 9 MONTHS AFTER BASELINING

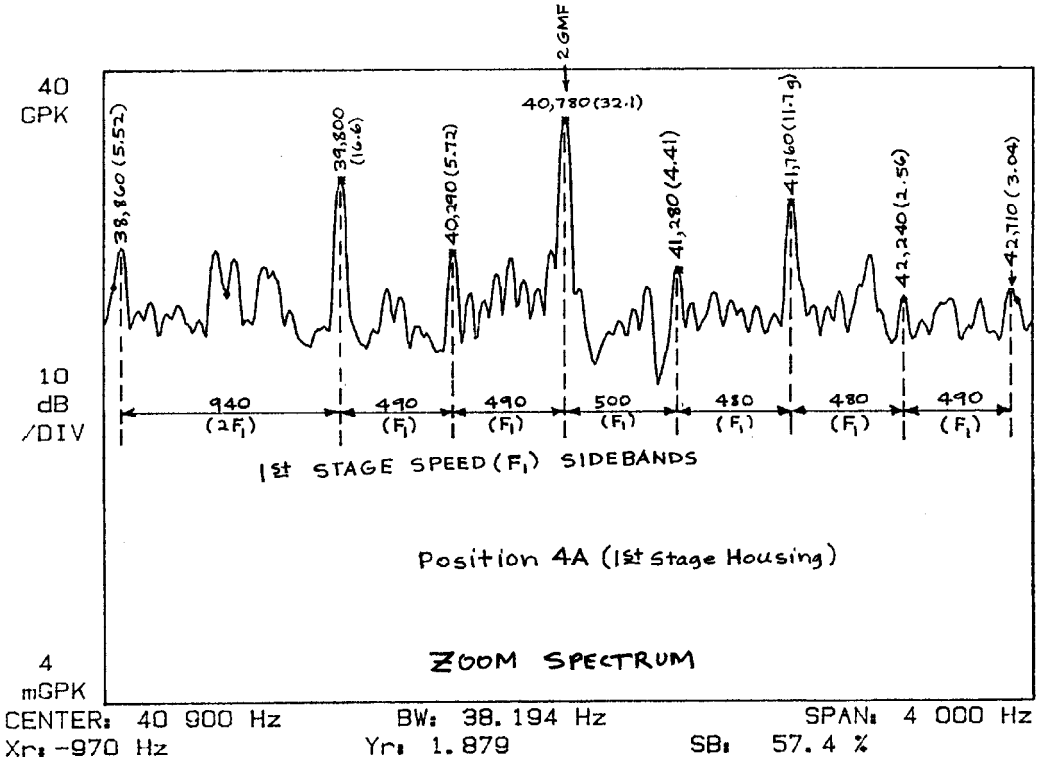

FIGURE 7: FIRST STAGE SPEED SIDEBANDS AROUND 2X GEAR MESH FREQUENCY

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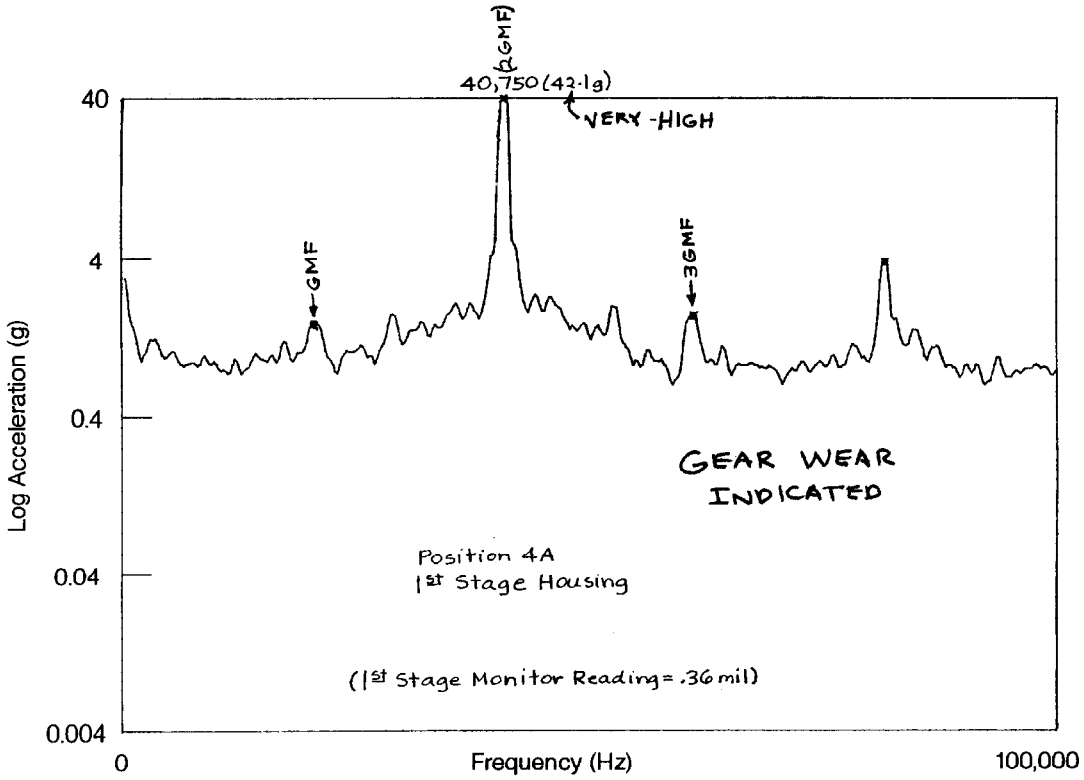

FIGURE 6: HIGH ACCELERATION DUE TO GEAR WEAR 9 MONTHS AFTER BASELINING

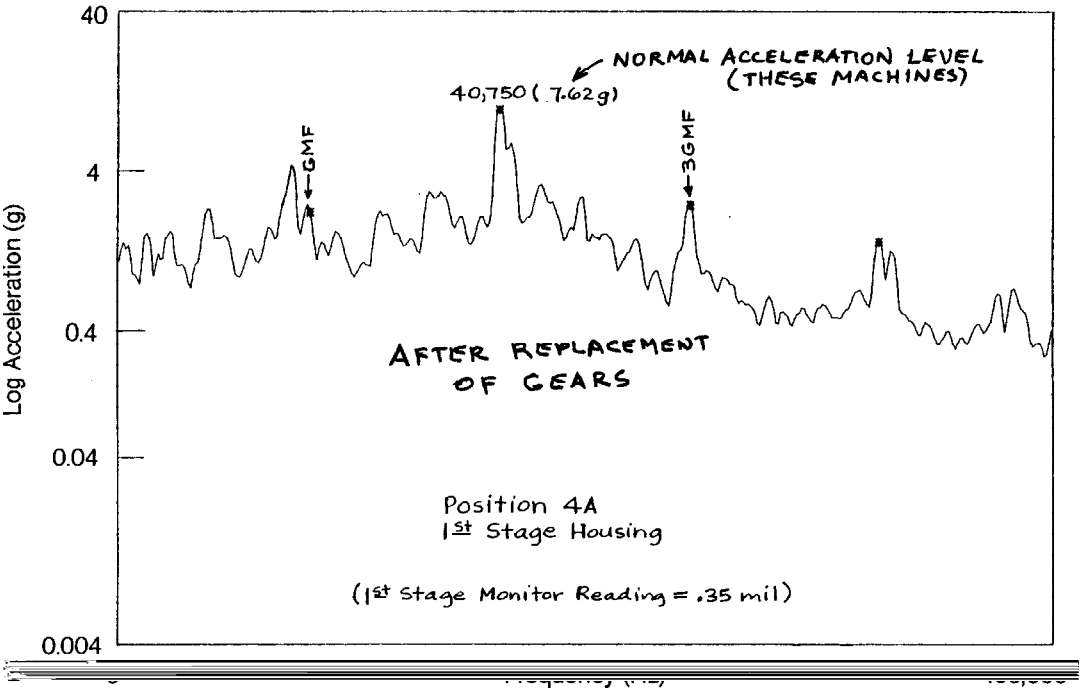

FIGURE 8: SPECTRUM AFTER REPLACEMENT OF BULLGEAR 1ST, 3RD & 4TH STAGE PINIONS

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DETECTION OF A SERIOUS ROLLING ELEMENT BEARING FAULT ON A SULLAIR SCREW COMPRESSOR

I. BACKGROUND

Technical Associates of Charlotte was required to provide vibration analysis diagnostic services on a Sullair screw compressor that appeared to be vibrating excessively. The testing was performed on June 24, 1991.

The Model 24K Sullair screw compressor was equipped with a 100 HP Reliance motor, which had been sent out for repair 3 times. The motor repair shop indicated that there was no problem with the motor this last time, but the unit continued to vibrate. The Sullair service representative recommended compressor bearing replacement. However, the client desired an independent verification of this diagnosis before work began. Measurements were also made on an identical compressor for comparative purposes as an aid in the diagnostic procedure. Figure 1 shows the compressor with measurement locations.

II. RESULTS AND CONCLUSIONS

A. METHODS

Technical Associates used a Rockland Model 5815A Real-Time Analyzer (RTA) in conjunction with a PCB Model 302 accelerometer to obtain the vibration data used for this diagnosis. A Monarch Model SPS-5 photocell was also available for use in obtaining phase measurements, but was not found to be necessary. Vibration data was obtained in three mutually perpendicular axes at each bearing location for the machines analyzed. The vibration frequency spectra were examined for indications of electrical problems, bearing wear, imbalance, misalignment, bent or eccentric rotors, resonances, rotor rubs, mechanical looseness and aerodynamic forces. The zoom capabilities of the RTA allow for detailed analysis of each frequency of interest.

B. ANALYSIS

1. Motor Condition

As the zoom spectrum in Figure 2 indicates, the motor speed on the screw compressor at the time of the test was 1774.5 RPM. An examination of all the motor vibration spectra indicates a maximum 1X RPM vibration amplitude of .279 in/sec at the same position illustrated in Figure 2 (motor, outboard bearing, horizontal direction). The alarm level for 1X RPM frequencies due to imbalance or any of the other forces which influence 1X RPM frequencies is generally considered to be 90% of the overall vibration alarm for any particular machine (see Table I). The 1X RPM vibration amplitude on the screw compressor of .279 in/sec is well below the alarm value of .383 in/sec (.425 X .90 = .383), indicating none of these forces acting singularly, or in combination were a cause for concern on this particular motor. Table II tabulates the vibration measured at each position shown on Figure 1.

Motor electrical problems are often indicated by excessive vibration amplitudes at 2X electrical line frequency (7200 CPM, or $2F_L$) and the amplitude of its related sidebands (if any are present). The zoom spectrum in Figure 3 indicates a very low vibration amplitude of .0079 in/sec at $2F_L$. The $4X\ F_M$ peak in Figure 3 was most likely transmitted from the compressor since one of the compressor rotors has 4 lobes. This frequency is called the "lobe mesh" frequency (F_{LM}). Motor vibration spectra do not indicate the presence of any other motor/electrical anomalies (such as stator eccentricity; shorted laminations; loose iron; eccentric rotors; broken, cracked rotor

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bars, shorting rings, shorted rotor laminations, or loose rotor bars; thermal bow; electrical phasing problems and torque pulse problems).

As Figure 4 indicates, the only other frequency prominent in the motor vibration spectra was a peak at 11,399 CPM. This peak was prominent throughout the entire machine and, as Table II indicates, became the dominant frequency throughout compressor vibration spectra. The measurement positions are illustrated in Figure 1. The 11,399 CPM peak (actually found to be 11,385 CPM in the Figure 5 zoom spectrum) was identified as a compressor bearing defect frequency. Neither Figure 4 nor any of the other motor vibration spectra give any indications of mechanical looseness or misalignment/coupling problems.

2. Compressor Condition

As stated in the previous section, a vibration amplitude peak at approximately 11,400 CPM was found to dominate all the compressor vibration spectra. Figure 5 is a zoom spectrum that more accurately defines the peak's frequency as 11,385 CPM. Table II indicates that out of 7 compressor vibration measurement positions where the overall vibration level exceeded .400 in/sec, 6 were axial direction readings and were dominated by the 11,385 CPM frequency. The vibration spectrum from the measurement position with the highest amplitude (Position 8, axial) is shown in Figure 6. This plot taken in the axial direction on the inboard bearing of the slow speed shaft (6 lobe, 1170 RPM) clearly indicates a defect in the SKF 7314 bearing outer race as evidenced by excessive vibration of .629 in/sec at 11,399 CPM (2X BPFO) and .267 in/sec at 17,100 CPM (3X BPFO). Table III contains the frequencies for an SKF 7314 bearing at 1170 RPM. The spectrum contains several multiples of the outer race defect frequency which normally indicates propagation of defects around the periphery of the bearing. In fact, these multiples of SKF 7314 outer race defect frequencies were the only abnormal frequencies appearing in the machine's vibration spectra. As a comparison to a machine operating normally, Figures 7 and 8 are included. These spectra have 30,000 CPM maximum frequencies (Figure 6 has 60,000 CPM maximum frequency), and are both measured at the same location as Figure 6 (Position 8-A). Figure 7 was measured on Sullair #1 screw compressor (defective) and Figure 8 was measured on Sullair #2 screw compressor (normal). Please note the extreme difference in full scale amplitude on the comparison plots, further indicating the great degree of degradation of Sullair #1's bearing. Also note the extreme difference in amplitude of the prominent frequency peaks listed on the comparison plots. The ability to identify several multiples of the bearing defect frequency in the vibration spectra indicates that the bearing was in the fourth and final stage of bearing failure. In this stage, the bearing was approaching catastrophic failure. Its remaining life was likely somewhere between 1 hour and 1% of its design life. Removing the bearing at this point was mandatory, or rotor/housing damage may have occurred during a catastrophic failure.

III. RECOMMENDATIONS

A. It was recommended that the compressor be overhauled immediately, replacing all bearings, seals, etc. Also, the client was told to closely inspect all compressor lobe and inner wall surfaces for any signs of possible rub/contact.

B. It was likewise recommended this compressor be placed in a regularly scheduled condition monitoring program to not only prevent potential catastrophic failure, but also to detect such problems long before they became this severe.

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IV. RESULTS OF CORRECTIVE ACTIONS

The Sullair No. 1 Screw Compressor was tested following completion of the overhaul. Figures 9 and 10 are included as a before/after comparison. As these plots indicate, the bearing vibration was completely absent! Also, amplitudes at input compressor RPM (F_M) and lobe mesh frequency (F_{LM}) were now well within acceptance criteria. Inspection of the compressor SKF 7314 bearing revealed excessive wear throughout the periphery of the outer race.

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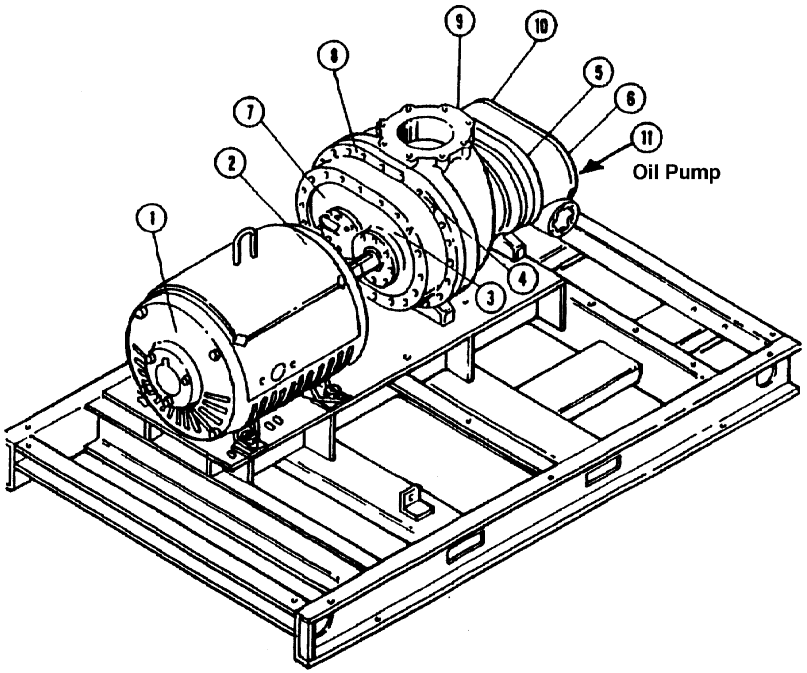

FIGURE 1
SULLAIR SCREW COMPRESSORS
MODEL 24KT

TABLE 1.
CRITERIA FOR OVERALL CONDITION RATING (PEAK OVERALL VELOCITY, IN/SEC)

1. Assuming Machine Speed = 500 to 60,000 RPM
2. Assuming Measurements by Accelerometer or Velocity Pickup as Close as Possible to Bearing Housing
3. Assuming Machine Not Mounted on Vibration Isolators (for isolated Machinery - Set Alarm 30% - 50% Higher)
4. Set Motor Alarms Same as that for the Particular Machine Type unless Otherwise Noted
5. Set Alarms on Individual External Gearbox 25% Higher than that for a particular Machine Type.

MACHINE TYPE	GOOD	FAIR	ALARM
COOLING TOWER DRIVES			
Long, Hollow Drive Shaft	0 - .375	.375 - .600	.600
Close Coupled Belt Drive	0 - .275	.275 - .425	.425
Close Coupled Direct Drive	0 - .200	.200 - .300	.300
COMPRESSORS			
Reciprocating	0 - .325	.325 - .500	.500
Rotary Screw	0 - .275	.275 - .425	.425
Centrifugal With or W/O External Gearbox	0 - .200	.200 - .300	.300
Centrifugal - Integral Gear (Axial Meas.)	0 - .200	.200 - .300	.300
Centrifugal - Integral Gear (Radial Meas.)	0 - .150	.150 - .250	.250
BLOWERS (FANS)			
Lobe-Type Rotary	0 - .300	.300 - .450	.450
Belt-Driven Blower	0 - .275	.275 - .425	.425
General Direct Drive Fans	0 - .250	.250 - .375	.375
Primary Air Fans	0 - .250	.250 - .375	.375
Large Forced Draft Fans	0 - .200	.200 - .300	.300
Large Induced Draft Fans	0 - .175	.175 - .275	.275
Shaft-Mounted Integral Fan	0 - .175	.175 - .275	.275
Vane-Axial Fan	0 - .150	.150 - .250	.250
MOTOR/GENERATOR SETS			
Belt-Driven	0 - .275	.275 - .425	.425
Direct Coupled	0 - .200	.200 - .300	.300
CHILLERS			
Reciprocating	0 - .250	.250 - .400	.400
Centrifugal (Open-Air)	0 - .200	.200 - .300	.300
Centrifugal (Hermetic)	0 - .150	.150 - .225	.225
LARGE TURBINE/GENERATORS			
3600 RPM Turbine/Geaerators	0 - .250	.250 - .375	.375
1800 RPM Turbine/Generators	0 - .175	.175 - .275	.275
CENTRIFUGAL PUMPS			
Vertical Pumps (12' - 20' Height)	0 - .375	.375 - .600	.600
Vertical Pumps (8' - 12' Height)	0 - .325	.325 - .500	.500
Vertical Pumps (5' - 8' Height)	0 - .250	.250 - .400	.400
Vertical Pumps (0' - 5' Height)	0 - .200	.200 - .300	.300
General Purpose Horizontal	0 - .200	.200 - .300	.300
Boiler Feed Pumps	0 - .200	.200 - .300	.300
Hydraulic Pumps	0 - .125	.125 - .200	.200
MACHINE TOOLS			
Motor	0 - .100	.100 - .175	.175
Gearbox Input	0 - .150	.150 - .225	.225
Gearbox Output	0 - .100	.100 - .175	.175
Spindles			
a. Roughing Operations	0 - .075	.075 - .125	.125
b. Machine Finishing	0 - .050	.050 - .075	.075
c. Critical Finishing	0 - .030	.030 - .050	.050

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SULLAIR #1 SCREW COMPRESSOR VIBRATION AMPLITUDES
(IN/SEC. PK)

POS/DIR	F_n	SKF 7314 BPFO	2X BPFO	3X BPFO	F_{LM}	OA LEVEL	EXCEEDS ALARM	COMMENTS
1A	.178	---	.036	---	.032	.194		OA alarm for this machine: .425 in/sec
H	.284	---	.059	---	.026	.309		
V	.161	---	.032	---	.026	.176		
2A	.200	---	.062	---	.022	.220		
H	.277	---	.062	---	.040	.301		
V	.115	.038	.029	---	.035	.148		
3A	.116	.058	.344	.103	.042	.482	X	2 BPFO (7314)
H	.093	.048	.225	.125	.021	.330		
V	.205	.053	.118	.035	.031	.318		
4A	.109	.090	.331	.067	.075	.475		
H	.087	.035	.172	.065	.023	.242		
V	.214	.056	.090	.018	.019	.304		
5A	.097	.084	.249	.058	.074	.342		
H	.056	.123	.048	.075	.056	.247		
V	.156	.075	.047	.026	---	.244		
6A	.101	.054	.269	.067	.043	.341		
H	.069	.168	.067	.093	.057	.289		
V	.183	.074	.026	.017	---	.243		
7A	.145	.041	.553	.292	---	.718	X	2 BPFO (7314)
H	.054	---	.313	.054	---	.347		
V	.199	.044	---	.023	---	.282		
8A	.155	---	.629	.267	---	.763	X	2 BPFO (7314)
H	.047	---	.172	---	.019	.211		
V	.209	.041	.039	.017	.032	.293		
9A	.158	.052	.501	.152	---	.605	X	2 BPFO (7314)
H	.037	---	.110	.062	.045	.218		
V	.207	.071	.022	.008	---	.276		
10A	.168	.052	.333	.110	---	.434	X	2 BPFO (7314)
H	.034	---	.176	.095	.050	.283		
V	.218	.056	.059	---	---	.289		
11A	.103	.037	.148	.067	---	.274		
H	---	.112	.388	.151	---	.563	X	2 BPFO (7314)

TABLE II

Bearing: SKF 7314 RPM: 1170

Multiple	Bpfo:	Bpfi:	Bsf:	Btf:
1X	5692	8348	2443	474
2X	11384	16696	4886	948
3X	17076	25044	7329	1422
4X	22768	33392	9772	1895
5X	28460	41740	12215	2369
6X	34152	50088	14658	2843

Bearing Number	Balls	BpfoX	BpfiX	BsfX	BtfX
7306	12	4.815	7.185	1.992	0.401
7307	12	4.847	7.153	2.052	0.404
7308	12	4.861	7.139	2.080	0.405
7309	12	4.817	7.183	1.996	0.401
7310	12	4.829	7.171	2.019	0.402
7311	12	4.841	7.159	2.042	0.403
7312	12	4.850	7.150	2.059	0.404
7313	12	4.859	7.141	2.075	0.405
7314	12	4.865	7.135	2.088	0.405
7315	12	4.872	7.128	2.101	0.406
7316	12	4.876	7.124	2.110	0.406
7317	12	4.882	7.118	2.121	0.407

TABLE III
BEARING DEFECT FREQUENCY INFORMATION

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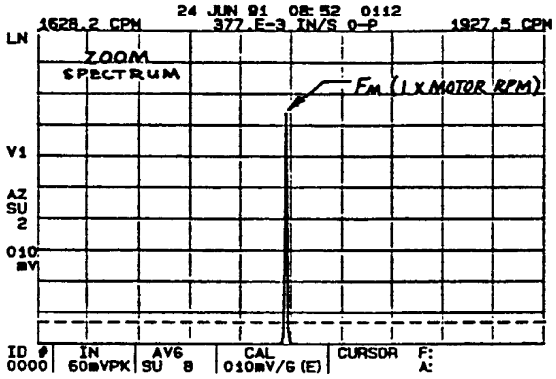


IN/S 0-P		CPM
279.E-3	F_M	1774.5

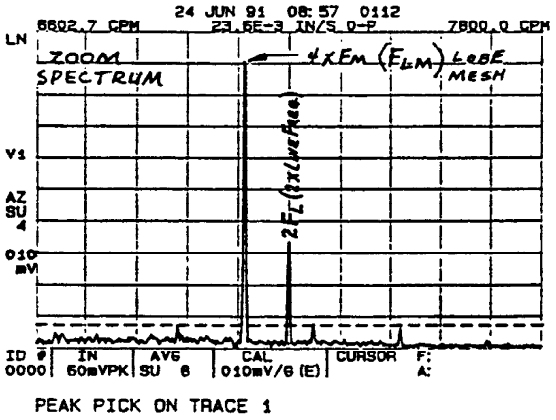
25-JUN-91 *MOTOR, OUTBOARD BEARING*

POSITION 1H

17: 31: 18 *HORIZONTAL DIRECTION*

SULLAIR #1

FIGURE 2



IN/S 0-P		CPM
1.62E-3		6935.7
21.4E-3	$4 \times F_M (F_{LM})$	7094.7
7.87E-3	$2F_L$	7200.0
1.72E-3		7257.0

25-JUN-91 *MOTOR, OUTBOARD BEARING*

POSITION 1H

17: 31: 53 *HORIZONTAL DIRECTION*

FIGURE 3

SULLAIR #1

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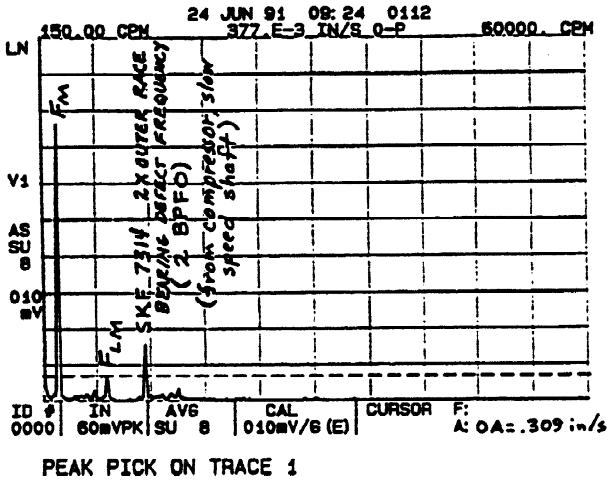


PEAK PICK ON TRACE 1

IN/S 0-P		CPM
289.E-3	F_M	1800.0
25.8E-3	F_{LM}	7050.0
58.5E-3	SKF 7314 2 BPFO (from compressor)	11399.

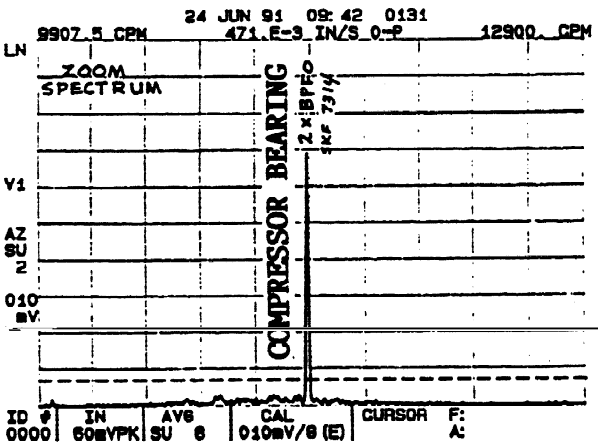
25-JUN-91 MOTOR, OUTBOARD BEARING

POSITION 1H

17: 44: 40 HORIZONTAL DIRECTION

FIGURE 4

SULLAIR #1



PEAK PICK ON TRACE 1

IN/S 0-P		CPM
327.E-3	SKF 7314 2 x BPFO (low speed shaft)	11385.

25-JUN-91 COMPRESSOR, DRIVE END (HIGH SPEED SHAFT)

POSITION 3A

17: 58: 27 AXIAL DIRECTION

FIGURE 5

SULLAIR #1

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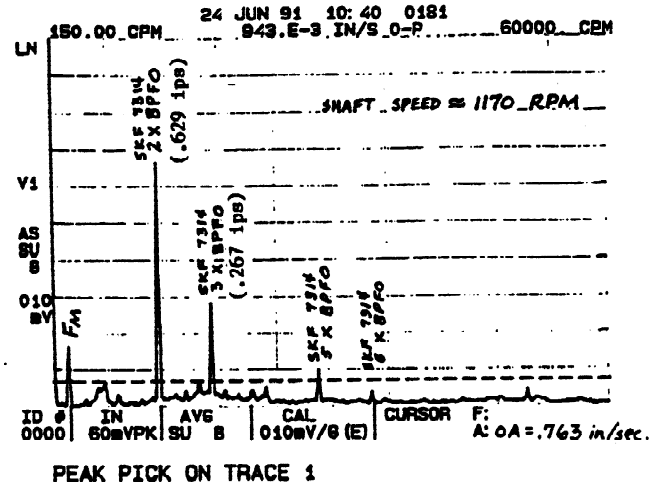

PEAK PICK ON TRACE 1

IN/S 0-P		CPM
155.E-3	F_M	1800.0
629.E-3	SKF 7314 2 X BPFO	11399.
64.6E-3		15599.
267.E-3	SKF 7314 3 X BPFO	17100.
90.4E-3	SKF 7314 5 X BPFO	28499.

25-JUN-91 COMPRESSOR, INBOARD (SLOW SPEED SHAFT)

19: 14: 51 AXIAL DIRECTION (POSITION 8-A)

SULLAIR #1

FIGURE 6

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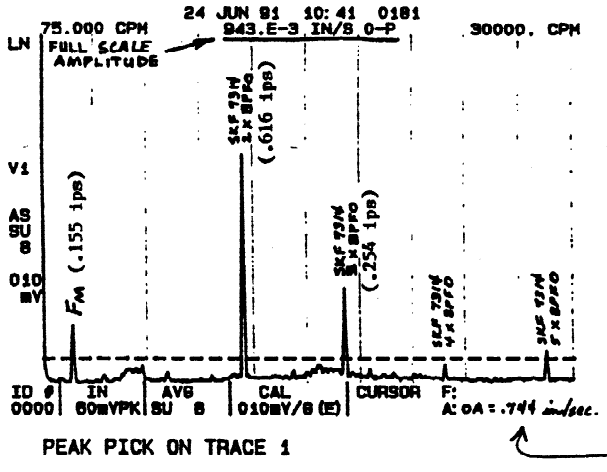


IN/S 0-P		CPM
155.E-3	F_M	1800.0
616.E-3	SKF 7314 2 x BPFO	11399.
254.E-3	SKF 7314 3 x BPFO	17100.
86.1E-3	SKF 7314 5 x BPFO	28500.

DEFECTIVE MACHINE

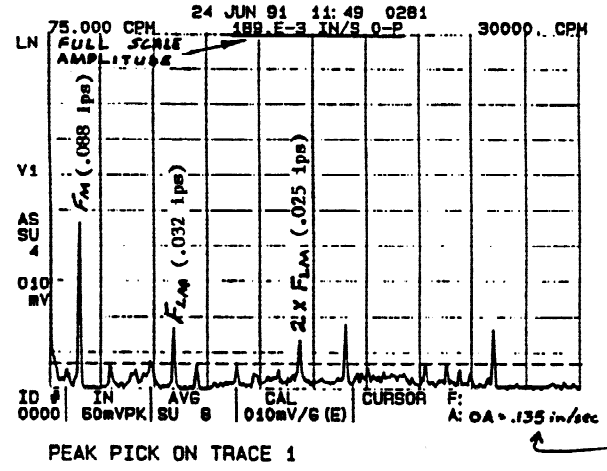
EXCESSIVE OVERALL AMPLITUDE

25-JUN-91 POSITION 8 AXIAL

19: 16: 17

FIGURE 7

SULLAIR #1



IN/S 0-P		CPM
88.0E-3	F_M	1800.0
13.8E-3		5775.0
31.9E-3	F_{LM}	7124.7
25.0E-3	2 x F_{LM}	14249.
33.7E-3		16800.
12.9E-3		21300.
31.1E-3		25200.

GOOD MACHINE

Good OVERALL AMPLITUDE

26-JUN-91 POSITION 8 AXIAL

07: 34: 22

FIGURE 8

SULLAIR #2

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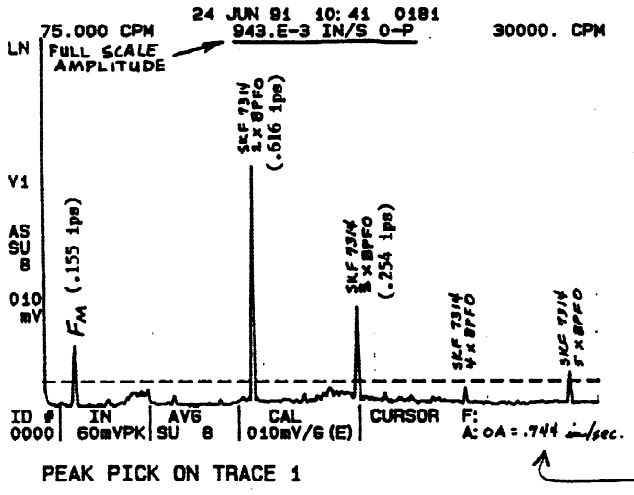


IN/S 0-P		CPM
155.E-3	FM	1800.0
616.E-3	SKF 7314 1 x BPFO	11399.
254.E-3	SKF 7314 2 x BPFO	17100.
86.1E-3	SKF 7314 5 x BPFO	28500.

BEFORE REPAIR

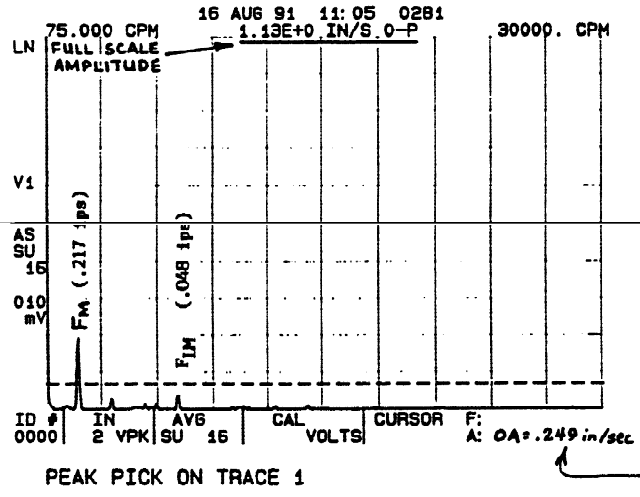
EXCESSIVE OVERALL AMPLITUDE

SULLAIR #1

25-JUN-91 POSITION 8 AXIAL

19: 16: 17

FIGURE 9



IN/S 0-P		CPM
217.E-3	FM	1800.0

AFTER REPAIR

ACCEPTABLE OVERALL AMPLITUDE

SULLAIR #1

19-AUG-91 COMPRESSOR, INBOARD (SLOW SPEED SHAFT)

12: 32: 17 AXIAL DIRECTION (POSITION 8-A)

FIGURE 10

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SURPRISING DETECTION OF A ROLLING ELEMENT BEARING PROBLEM ON A DRAW FRAME INPUT PINION BEARING AFTER REPLACEMENT OF A FALK GEARBOX WHICH HAD BEEN THOUGHT TO BE THE SOURCE OF THE PROBLEM

Subtitle:
(How it Pays to Know the Identity of Components Within Your Critical Machinery)

I. BACKGROUND

Technical Associates of Charlotte, P.C. was contacted with regards to performing **Vibration Diagnostic Testing** on a Draw Frame gearbox shown in Figure 1. The problem was described as a bumping noise and vibration that appeared to be occurring at once per revolution of the gearbox output speed. This draw frame includes each of 30 rolls of 18 inch diameter supported on 9 inch diameter shafts as shown in Figure 2. These rolls are gearbox driven and are outfitted with high cost bearings. There are several individual drives for various sections of the 30 draw frame rolls including one like that shown in Figure 1. This draw frame is used in production of a high strength fiber material called staple ranging in density from 2,000,000 to 3,000,000 denier. From Figure 2, the tow band applies very high forces on the order of 30,000 lbs on each of the bearings supporting the draw frame rolls as the staple wraps around some rolls greater than 180 degrees. This can put great strain on many of the draw frame components as well as their drives.

The Falk gearbox shown in Figure 1 had been replaced at a cost of over $36,000 which unfortunately had little or no effect on the high vibration problem. Several draw frame roll bearings had also been replaced which likewise showed no improvement. Therefore, the plant requested outside assistance to help identify just what was the source of the problem and what was required to resolve it.

The **Vibration Diagnostic Testing** was performed on June 16, 1989.

The following report contains CONCLUSIONS, RECOMMENDATIONS, and a DISCUSSION of test data obtained.

II. CONCLUSIONS

A. Surprisingly, the problem in this machine had nothing to do with components in either the gearbox nor motor, but instead with the bearing supporting the input pinion of the draw frame itself (see Figure 1). An inner race defect frequency and harmonics for the FAG Model #22326ES bearing was detected. This is the bearing closest to the draw frame input pinion. The amplitudes and frequencies associated with the bearing defect indicated that the bearing was in the final stages of failure.

B. Once the data on bearing model numbers and gear tooth counts was made available, the analysis was straightforward, although the vibration spectra had initially appeared very strange when such information was unknown.

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C. Data obtained showed both the gearbox and motor to be in good condition.

D. Due to the inaccessibility of the large draw frame roll bearings mounted deep within the machine, the mechanical condition of these bearings could not be assessed. Recommendations were made by the analyst while on-site how this situation could be remedied by placing permanently mounted accelerometers in key locations which were identified on machine prints given to the analyst. It was important to take the speed of these rolls into account when specifying both hardware and techniques since they normally ranged from only 32 to 104 RPM (see Figure 2).

III. RECOMMENDATIONS

A. Recommendations were made while on-site to replace the Fag 22326ES Draw Frame input pinion bearings as soon as possible to prevent catastrophic failure which may have been imminent as evidenced by vibration spectra.

B. Monitor the gearbox utilizing the client's existing IRD 890/7090 **Predictive Maintenance Program** hardware and software to detect similar problems in the future. Also, complete the research on identification of bearing and gear components within the draw frame as well as its driving gearboxes and motors. This data could prove to be invaluable preventing not only future failures of these expensive components, but also helping to prevent costly expenditures replacing gearboxes and/or motors that actually are in good operating condition.

IV. DISCUSSION

A. Initial Analysis Without Available Information on Motor, Gearbox and Draw Frame Components:

Upon arrival at the plant, information concerning the motor, gearbox and draw frame bearing model numbers and gear tooth counts was not yet available. However, after capturing several interesting, high amplitude spectra like those shown in Plots 1, 2 and 3, it became mandatory that this information be supplied since the sources of most of the peaks could not be identified.

Plot 1 showed three peaks, only one of which could be identified due to the lack of information on components within the motor, gearbox and draw frame. Plot 1 is a zoom spectra (taken on position 1H) which included the motor speed of 1229.3 CPM. It also contained unidentified peaks at 1183.5 CPM and 1318.5 CPM. However, even though these two peaks were unidentified, it was noticed that their difference frequency was 135 CPM which equaled the output speed of the gearbox (1318.5-1183.5 = 135 CPM).

Data captured on the gearbox output shaft at positions 7A and 8A looked very unusual (see Plots 2 and 3). None of the dominant peaks which included frequencies of 1170, 2370 and 3540 CPM could be identified at that time. These three frequencies could not be related to either fundamentals nor harmonics of either the gearbox input speed (1229 RPM), intermediate speed (660 RPM) nor output speed (133 RPM). Also unusual about Plots 2 and 3 was a whole series of sideband frequencies which were spaced at gearbox output speed surrounding the 1170, 2370 and 3540 CPM peaks (as confirmed by zoom spectra taken around the approximate 1170 and 2370 CPM peaks shown in Plots 4 and 5). Depending on the cause of these spectra generating this unique pattern of frequencies, it was felt that the problem could in fact be quite severe due to amplitudes up to .351 in/sec and sideband levels up to almost .100 in/sec shown in Plots 4 and 5 (bearings, gears, electrical problems?). However, there simply was not sufficient machine component information available.

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Therefore, Technical Associates' analyst strongly asserted this data simply must be made available. Despite several arguments claiming this would require an extended time, the client finally dispatched a technician to research this information as quickly as possible.

B. Subsequent Analysis Once Gearbox Component Data Was Supplied:

Although it took approximately 2 hours, the plant finally was able to supply information on the gearbox, but advised no information was available at that time concerning components within either the motor nor draw frame.

The following mechanical frequencies were calculated based on the motor speed of 1229.3 RPM as seen in Plot 1 and the component information supplied (the speed was also confirmed by a strobe light):

Motor Speed		= 1, 229.3 RPM
Input Shaft Gear Mesh	1229.3(22T)	= 27, 044 CPM
Intermediate Shaft Speed	27,044.6/41T	= 659.62 RPM
Intermediate Pinion Gear Mesh	659.62(20T)	= 13,192.5 CPM
Output Shaft Speed	13,192.4/99T	= 133.25 RPM

GEARBOX BEARINGS

1.	Input Shaft Bearings	(Timken Nos. 6559 and 6536)	
	Outer Race (BPFO)	8.23 (1229.3)	= 10,117.14 CPM
	Inner Race (BPFI)	10.77 (1229.3)	= 13,239.56 CPM
	Ball/Roller (BSF)	7.14 (1229.3)	= 8,777.20 CPM
	Cage (FTF)	.43 (1229.3)	= 538.60 CPM
2.	Intermediate Shaft Bearings	(Timken Nos. HH224335 and HH224310)	
	Outer Race (BPFO)	7.11 (659.62)	= 4,689.90 CPM
	Inner Race (BPFI)	9.88 (659.62)	= 6,517.0 CPM
	Ball/Roller (BSF)	5.87 (659.62)	= 3,872.0 CPM
	Cage (FTF)	.42 (659.62)	= 277.0 CPM
3.	Output Shaft Bearings	(Timken No. 99100)	
	Outer Race (BPFO)	10.19 (133.25)	= 1,357.80 CPM
	Inner Race (BPFI)	12.81 (133.25)	= 1,707.0 CPM
	Ball/Roller (BSF)	8.40 (133.25)	= 1,119.30 CPM
	Cage (FTF)	.44 (133.25)	= 58.60 CPM

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Even after compiling all this data, the dominant peaks in spectra shown in Plots 1 through 5 still could not be pinpointed. Each of these spectra were dominated by a frequency of about 1183 CPM or multiples of this frequency. Therefore, information was again requested on the components within the motor and the draw frame which is driven by this gearbox. Several technicians now took approximately 4 hours to research this data. Finally, they were able to acquire information on the draw frame components. This finally uncovered the source of the real problem which was never within gearbox components at all. This will next be discussed.

SCHEMATIC VIEW OF LINES #17 AND #18 DRAW FRAMES

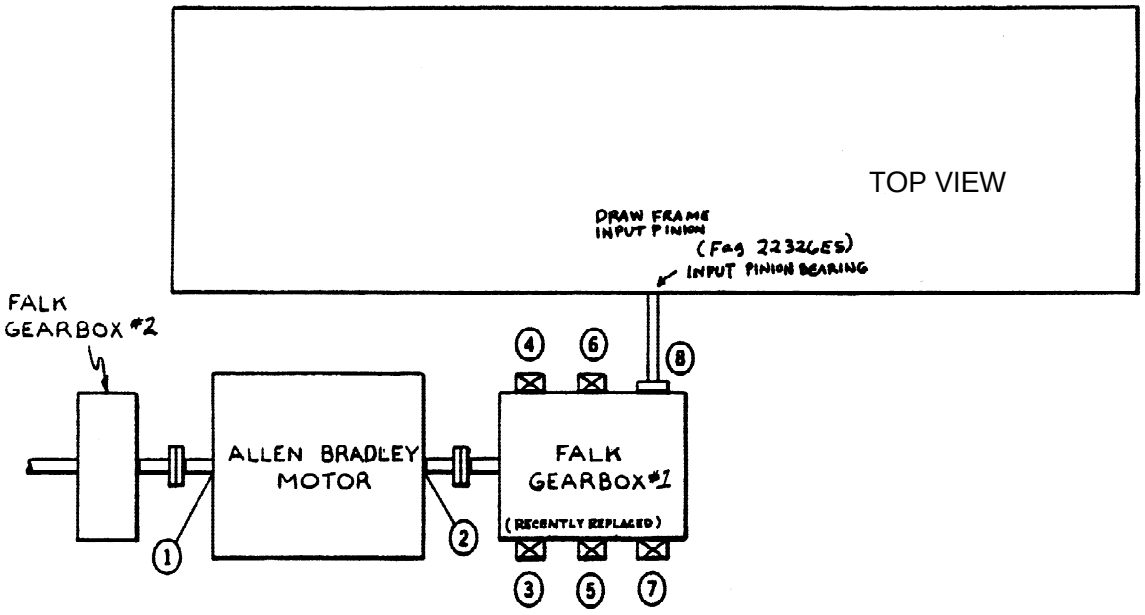

FIGURE 1

DETECTION OF A ROLLING ELEMENT BEARING PROBLEM IN A DRAW FRAME GEARBOX

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C. Results After Identification of Motor and Draw Frame Components:

Following below are calculated mechanical frequencies for the draw frame components once this information was finally made available:

DRAW FRAME PINION BEARINGS

1. Inboard Bearing, near Input Pinion (FAG #22326ES)

Outer Race	(BPFO)	6.12 (133.25) =	815.50 CPM	
Inner Race	(BPFI)	8.88 (133.25) =	1,183.3 CPM	← **Problem Source**
Ball/Roller	(BSF)	2.65 (133.25) =	353.10 CPM	
Cage	(FTF)	.41 (133.25) =	54.60 CPM	

2. Outboard Bearing (Timken #22226ES)

Outer Race	(BPFO)	8.22 (133.25) =	1,095.30 CPM
Inner Race	(BPFI)	10.80 (133.25) =	1,439.10 CPM
Ball/Roller	(BSF)	3.53 (133.25) =	470.40 CPM
Cage	(FTF)	.43 (133.25) =	57.30 CPM

Once the information shown above was made available, the solution was straightforward. Please refer to Plots 6 and 7 which are repeats of zoom Plots 4 and 5, but with each of the formerly unidentified peaks now pinpointed. Plots 6 and 7 are zoom spectra with a resolution of 3 CPM. Therefore, the peaks at 1185 and 2367 CPM were confirmed to be 1X BPFI and 2X BPFI for the FAG 22326ES draw frame input pinion bearing, respectively. Then, each of the 135 CPM sidebands surrounding them were found to be spaced at 1X RPM of the draw frame input pinion bearing.

There are normally four (4) stages of rolling element bearing failure as shown in Figure 3. In the early stages of bearing failure, the natural frequencies of bearing components (races, rolling elements, cages, etc.) are excited. These frequencies usually occur between 30,000 and 120,000 CPM. During later stages of failure, bearing defect frequencies appear. Then, as the defect propagates around the periphery of the bearing races, harmonics of defect frequencies appear and develop sidebands normally spaced at running speed when wear becomes clearly visible. In the final stage, 1X RPM will increase in amplitude and the defect frequencies will diminish or disappear completely. At this point, the bearing can fail at any time.

Since the bearing wear was obviously well into late stage 3, it was strongly felt that wear of this bearing was likely quite severe. This was particularly concerning when it was realized that the spectrum of Plot 1 was captured all the way back on position 1H on the motor outboard bearing; even here the dominant vibration was not at 1X motor RPM, but instead at 1X BPFI of the draw frame pinion bearing remotely located from this position (see Figure 1). Therefore, the analyst recommended replacing the bearing as soon as possible.

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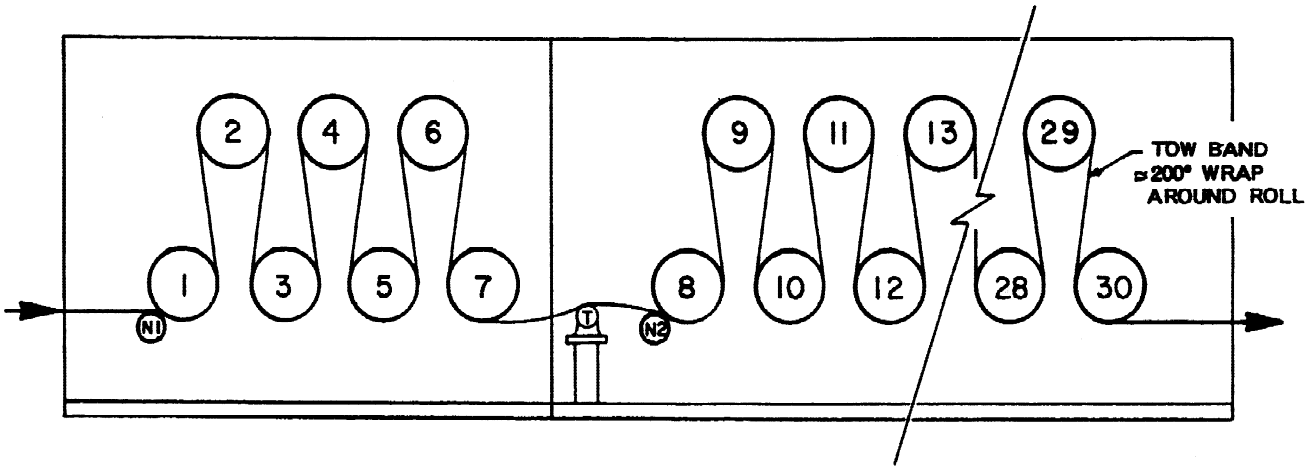
V. RESULTS OF CORRECTIVE ACTIONS

NOTE: Plot 8A was taken on the same machine at Gearbox Position 8A after replacement of the draw frame input pinion bearing (FAG 22326ES). It compares directly to Plot 3 taken before (which is shown below Plot 8A as “Plot 8B” to help the reader in spectral comparison). Note the significant drop in vibration and the total elimination of draw frame bearing defect frequencies. As of the writing of this case history almost 2 years since this diagnostic investigation, the draw frame has not experienced any further failures and continues to enjoy low vibration and noise levels.



ROLL INDENT	490 FPM (RPM)	158 FPM (RPM)	ROLL DIAMETER
1 - 5	31.7	10.2	18.0"
6	67.6	21.8	18.0"
7	94.6	30.5	18.0"
8 - 30	104.0	33.5	18.0"
NIP-1	71.3	23.0	8.0"
NIP-2	234.0	75.4	8.0"
TENSION	651.0	210.0	2.875"

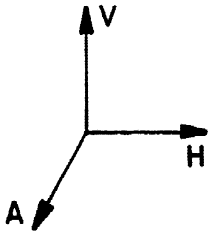


FIGURE 2
TENSION METER SET-UP ON DRAW FRAME

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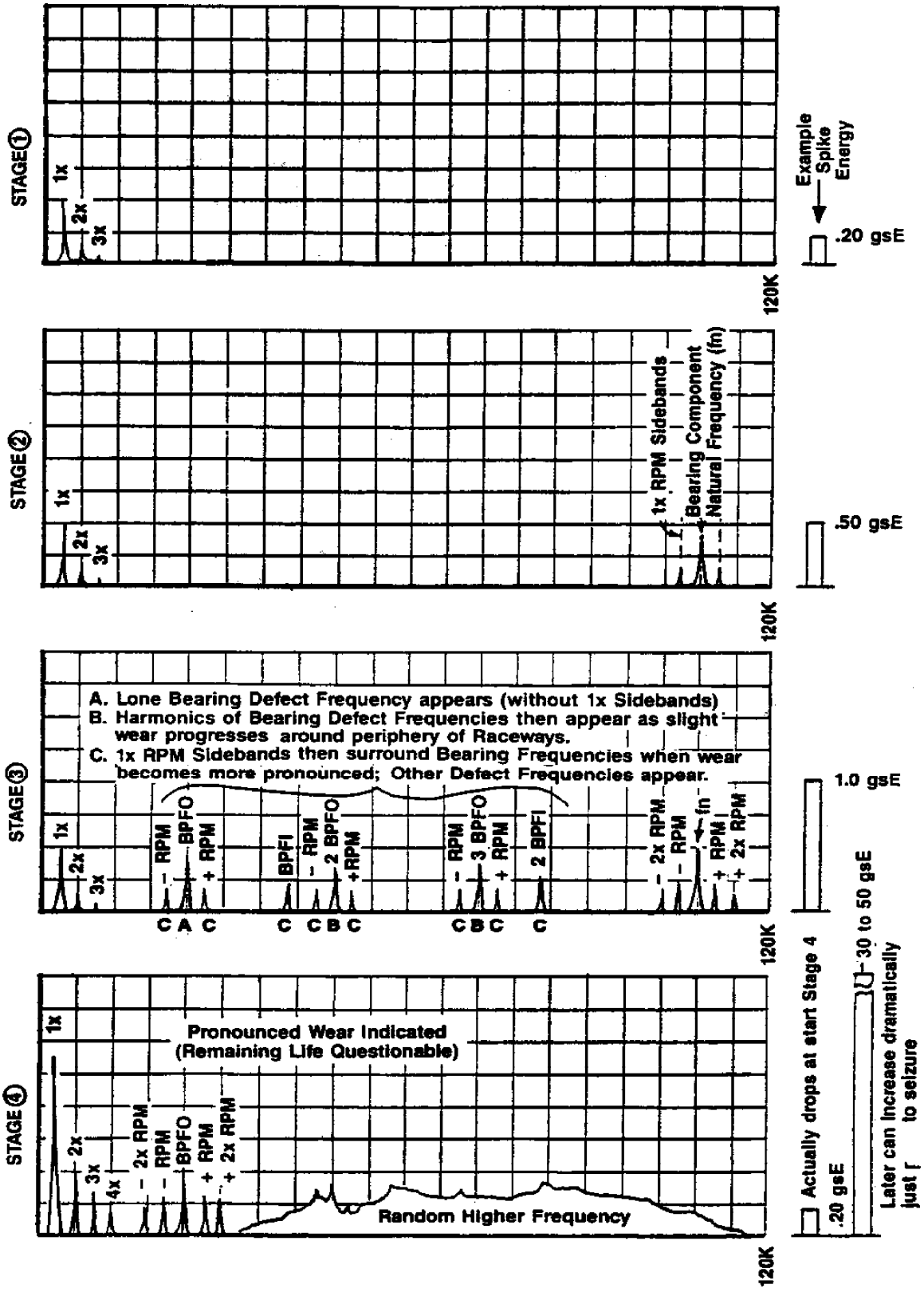
FIGURE 3
4 PRIMARY FAILURE STAGES THRU WHICH MOST ROLLING ELEMENT BEARINGS PASS (velocity spectra)



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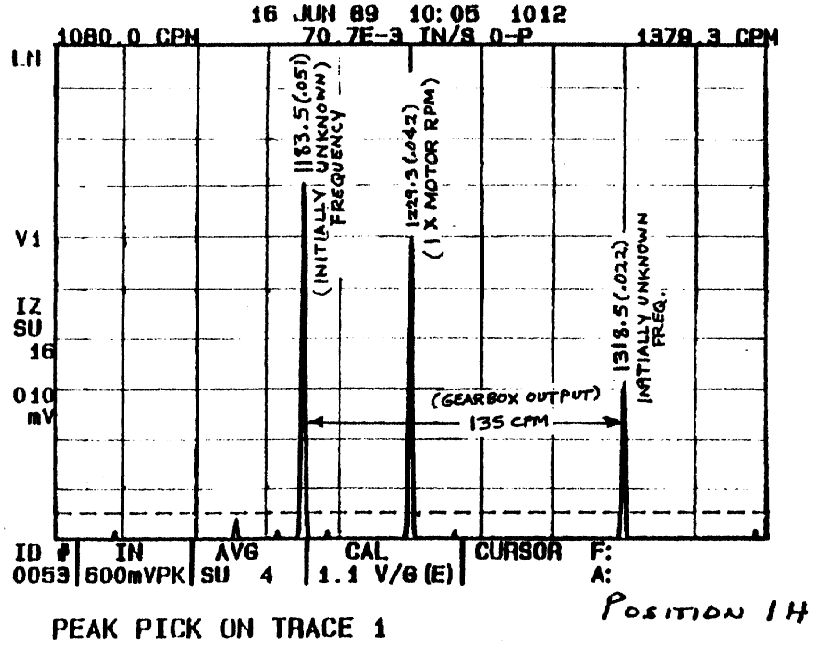

PEAK PICK ON TRACE 1

IN/S 0-P	CPM
51.3E-3	1183.5
42.3E-3	1229.3
21.6E-3	1318.5

21-JUN-89

07: 28: 04

PLOT 1

LINE 18

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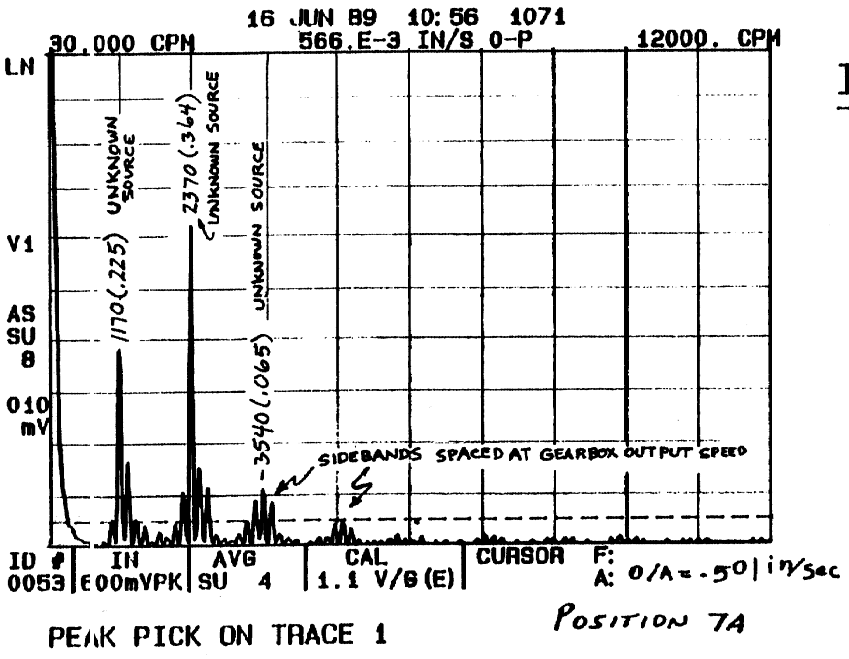

IN/S 0-P	CPM
31.0E-3	1050.0
* 225.E-3	1170.0
98.2E-3	1320.0
31.0E-3	1440.0
62.0E-3	2220.0
* 364.E-3	2370.0
90.4E-3	2490.0
67.2E-3	2640.0
28.4E-3	3269.9
51.7E-3	3420.0
64.6E-3	3540.0
49.1E-3	3690.0

21-JUN-89

07: 52: 41

PLOT 2

LINE 18

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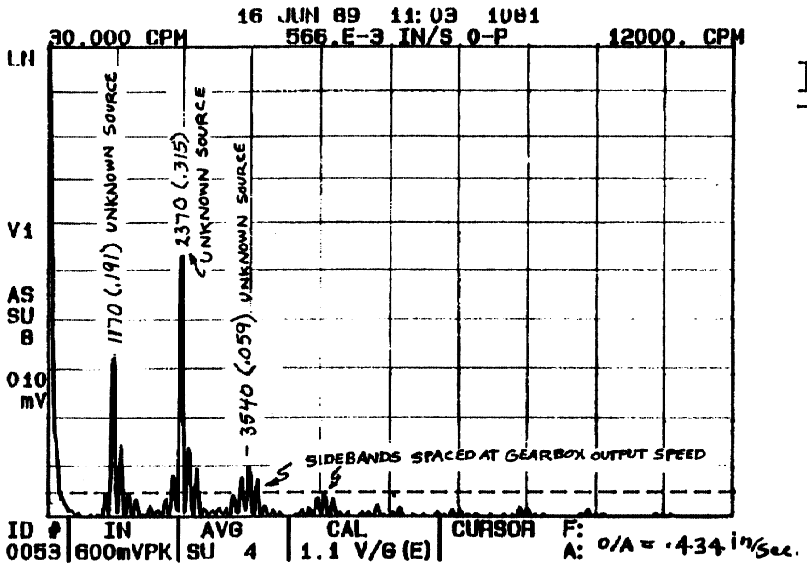

PEAK PICK ON TRACE 1

IN/S 0-P	CPM
*191.E-3	1170.0
85.3E-3	1320.0
49.1E-3	2220.0
*315.E-3	2370.0
82.7E-3	2490.0
56.9E-3	2640.0
46.5E-3	3420.0
59.4E-3	3540.0
43.9E-3	3690.0

21-JUN-89

08: 00: 03

LINE 18

PLOT 3

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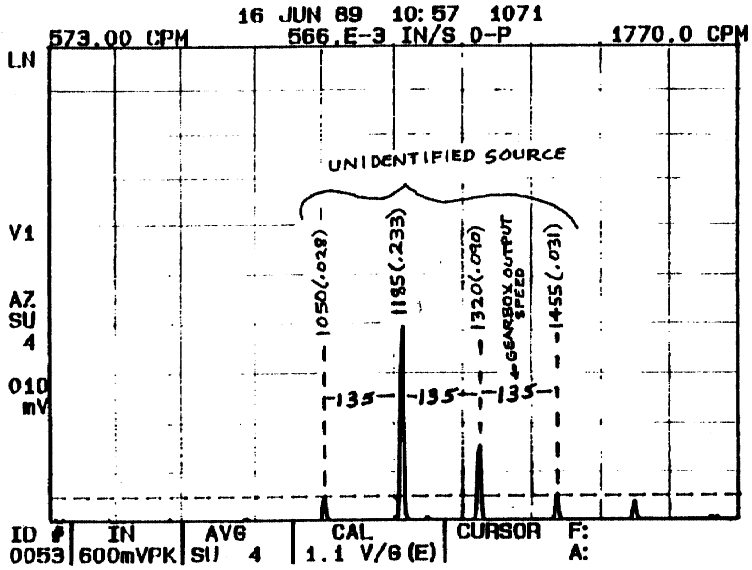

IN/S 0-P	CPM
28.4E-3	1050.0
233.E-3	1185.0
90.4E-3	1320.0
31.0E-3	1455.0

21-JUN-89

07:54:43

PLOT 4

LINE 1

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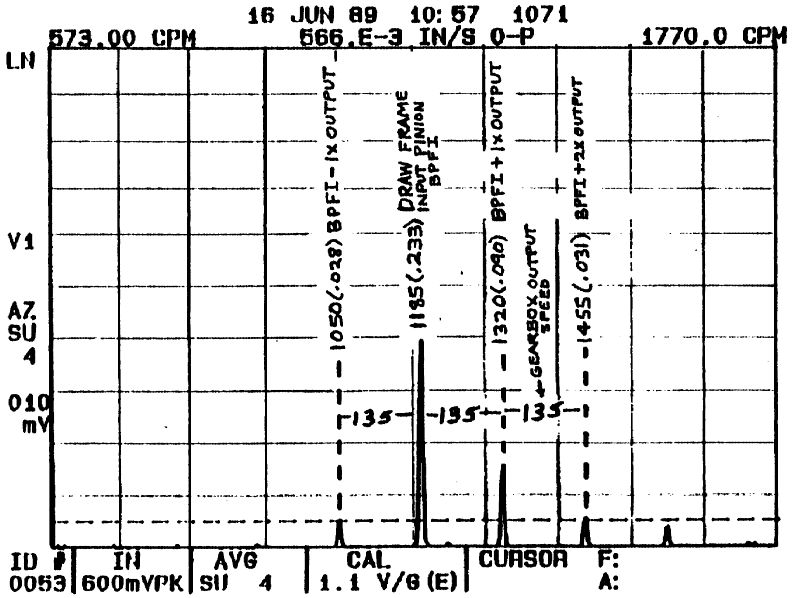

PEAK PICK ON TRACE 1

POSITION 7A

IN/S 0-P	CPM
28.4E-3	1050.0
233.E-3	1185.0
90.4E-3	1320.0
31.0E-3	1455.0

21-JUN-89

07:54:43

SIGNIFICANT WEAR INDICATED OF FAG 22326 ES DRAW FRAME INPUT PINION BEARING

PLOT 6

LINE 1

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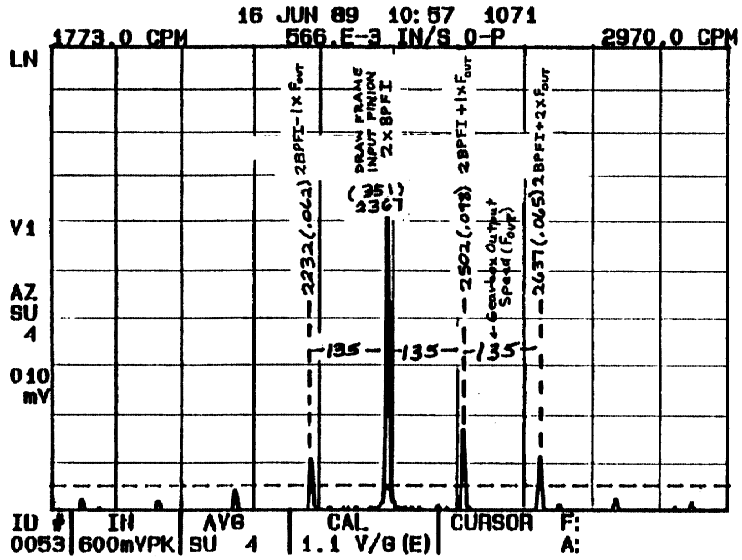


PEAK PICK ON TRACE 1

POSITION 7A

IN/S 0-P	CPM
62.0E-3	2232.0
351.E-3	2367.0
98.2E-3	2502.0
64.6E-3	2637.0

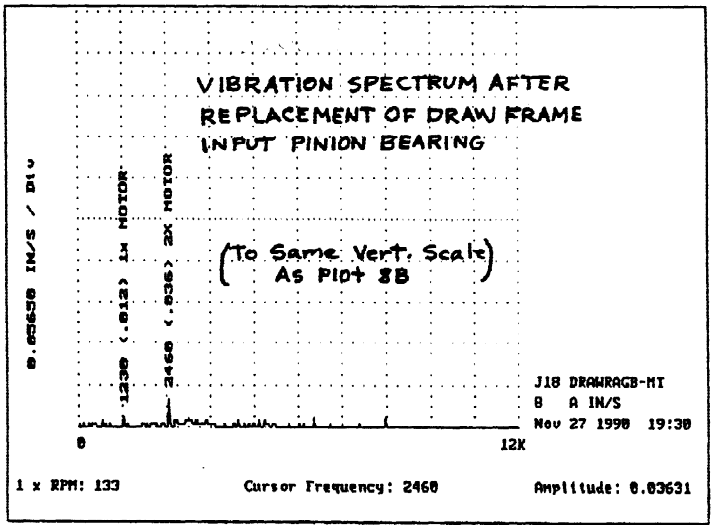
21-JUN-89

07: 56: 19

SIGNIFICANT WEAR INDICATED OF FAG 22326 ES DRAW FRAME INPUT PINION BEARING

PLOT 7

LINE 18



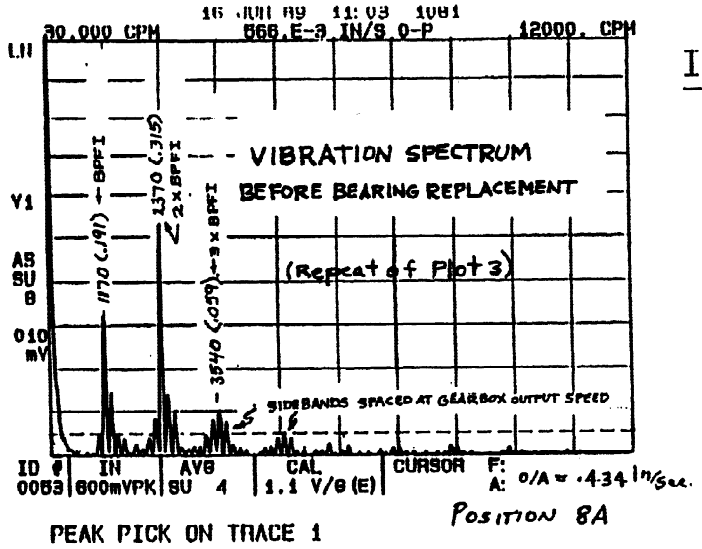
003% x 0.1000 = 0.00300 1 x RPM:

Machine	Pos	Dir	Units	Date
J18 DRAWRAGB-MT	8	A	IN/S	Nov 27 1990 19:30

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
390	-	0.00516	3510	-	0.00718
480	-	0.00405	3600	-	0.00356
660	-	0.00599	3690	-	0.00521
1230	-	0.01231	4230	-	0.00437
1740	-	0.00359	4350	-	0.00589
1980	-	0.00332	4770	-	0.00370
2310	-	0.00451	4920	-	0.00612
2460	-	0.03631	5100	-	0.00751
2670	-	0.00797	5340	-	0.00492
2790	-	0.00443	6150	-	0.00334
2910	-	0.00786	6420	-	0.00902
3000	-	0.01072	8370	-	0.01001
3180	-	0.00720			
3240	-	0.00600			
3330	-	0.00446			
3390	-	0.00426			
3450	-	0.00656			

Threshold adjust Q - Quit

PLOT 8A



IN/S 0-P	CPM
*191.E-3	1170.0
85.3E-3	1320.0
49.1E-3	2220.0
*315.E-3	2370.0
82.7E-3	2490.0
56.9E-3	2640.0
46.5E-3	3420.0
59.4E-3	3540.0
43.9E-3	3690.0

PLOT 8B

21-JUN-89

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SEVERE VIBRATION TREND OF A TIMKEN ROLLER BEARING INSTALLED IN A CIRCULATING FAN

I. BACKGROUND

F 21 Circulating Fan #4 suffered a severe bearing frequency vibration problem with its inboard fan bearing which is pictured as Position 3 Figure 1. Note that Timken model 34478 bearings are installed on the fan which is overhung from these two bearings. Severe vibration and spike energy levels were measured during the August 3, 1990 survey which mandated close analysis to prevent potentially catastrophic failure. This machine is analyzed on a regularly scheduled basis as part of a Predictive Maintenance Program annual contract.

II. ANALYSIS RESULTS

Figure 2 shows the vibration spectrum measured on Position 3A of the F 21 Circulating Fan #4 taken on August 3. Note that both Bands 5 and 6 were in alarm especially due to frequencies at 12,870 CPM (Band 5) and 25,740 CPM (Band 6). Comparison with the Timken 34478 bearing defect frequencies in Figure 1 confirmed that the frequency at 12,870 CPM was the outer race frequency (BPFO) while that at 25,740 CPM was its second harmonic (2X BPFO).

Figure 3 shows the trend of spike energy and that of the Band 6 velocity level between March 30, 1989 and October 11, 1990. Note the tremendous increase in spike energy from .630 to 2.89 gSE on August 3. Figure 4 shows two monthly measurement reports which displayed overall vibration at each point both before bearing replacement (8-3-90 report) and after bearing replacement (10-11-90 report). Note the significant decrease in both vibration and spike energy for fan Positions 3A, 3H and 3V which occurred on the October 11 report (decreased from 80% to 93%) after replacement of the fan bearings.

Figure 5 shows a waterfall plot which illustrates how the vibration spectrum at Position 3A changed from March 30, 1989 through October 11, 1990. Note that no bearing frequencies (or any high frequency energy for that matter), were present on either the March 30 or April 25 signatures. However, high amplitude outer race frequencies did appear as of the December 12 spectrum (.270 in/sec at 2X BPFO). Then, both BPFO and 2X BPFO roughly doubled as of the January 18, 1990 spectrum. They then continued to increase with the next spectrum taken August 3 up to the final levels of .150 in/sec at BPFO and an excessive .775 in/sec at 2X BPFO.

III. RECOMMENDATIONS MADE

Technical Associates recommended immediate replacement of the fan bearings in order to prevent potential catastrophic failure (recommendations have been made to replace these bearings since the January 18 spectrum was captured). In addition, it was recommended that the plant take steps to balance the fan which had trended from only about .120 in/sec at fan speed up to approximately .320 in/sec as of the August 3 spectrum.

IV. CORRECTIVE ACTIONS TAKEN

The plant did follow through with replacement of each of the fan bearings, but did not schedule the fan balancing at this time. Also, the plant took extra efforts to ensure good alignment between the motor and fan shafts.

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V. RESULTS OF CORRECTIVE ACTIONS

Figure 5 shows that all higher frequencies were eliminated in the spectrum of October 11 taken after bearing replacement. In addition, some improvement in fan vibration occurred even without the fan balance. Similarly, Figure 3 shows that both the spike energy and the overall velocity levels dropped significantly following bearing replacement.

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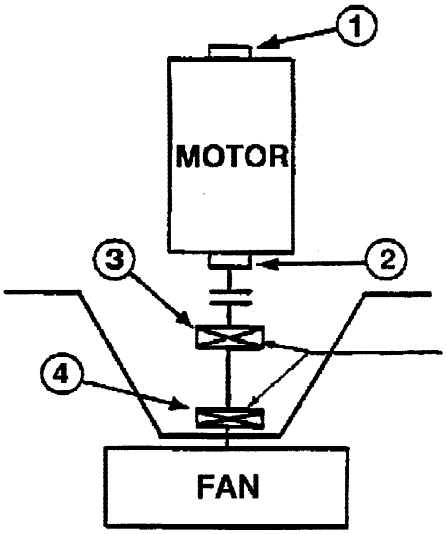
MACHINERY DATA GUIDE SHEET

RELIABILITY SURVEY predictive maintenance program (Page__of__)



TIMKEN 34478
BPFI = 13.211X RPM
BPFO = 10.789X RPM
BSF = 4.734X RPM
FTF = 0.450X RPM

MACHINE IDENT: FF 21 CIRCULATING FANS (1-6)

AREA:_____

ROUTE:_____

DRIVER	INTERMEDIATE	DRIVEN
RPM min : 1180 NAME PLATE	RPM in : _____	RPM min : 1180 RPM NAME PLATE
max : _____	out : _____	max : _____
MFGR : _____	RATIO: _____	MFGR. : _____
MODEL : _____	MODEL : _____	MODEL : _____
SER NO. : _____	SER NO. : _____	SER NO. : _____
V____ P____ A____	RATING : _____	RATING : _____
SHEAVE : _____ DIA.	BELT # : _____	#BLADES : _____
BRG#: _____ (I)	BRG#: _____ (I)	BRG#: TIMKEN 34478 (I)
BRG#: _____ (O)	BRG# _____ (O)	BRG#: TIMKEN 34478 (O)
H.P.: 40	GEAR T: _____ CT.	GEAR T: _____ CT.
OTHER: _____	OTHER: _____	OTHER: _____

DCB-8

FIGURE 1

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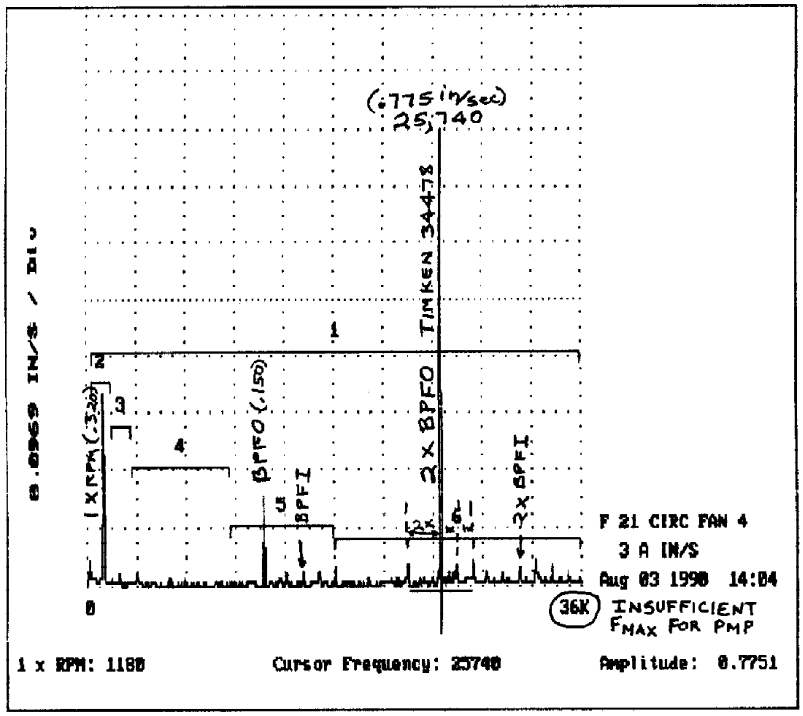

002% x 1.000 = 0.0200 1 x RPM: 1180.00

Machine	Pos	Dir	Units	Date
F 21 CIRC FAN 4	3	A	IN/S	Aug 03 1990 14:04

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
180	0.153	0.0450	31500	26.695	0.0300←2BPFI
1170	0.992	0.3199	32670	27.686	0.0450
2340	1.983	0.0200	33840	28.678	0.0349←2BPFI + 2X
3600	3.051	0.0200	35010	29.669	0.0249
12870	10.907	0.1500←BPFO			
14580	12.356	0.0200			
15750	13.347	0.0249← BPFI			
16920	14.339	0.0249			
18090	15.331	0.0300			
23400	19.831	0.0349← 2BPFO - 2X RPM			
25740	21.814	0.7751← 2X BPFO			
26370	22.347	0.0249			
26730	22.653	0.0249			
26910	22.805	0.0349← 2 BPFO + 1X RPM			
28080	23.797	0.0450← 2 BPFO + 2X RPM			
29070	24.636	0.0249← 2 BPFI - 2X RPM			
30330	25.703	0.0200← 2 BPFI - 1X RPM			

T - Threshold adjust Q - Quit

FIGURE 2

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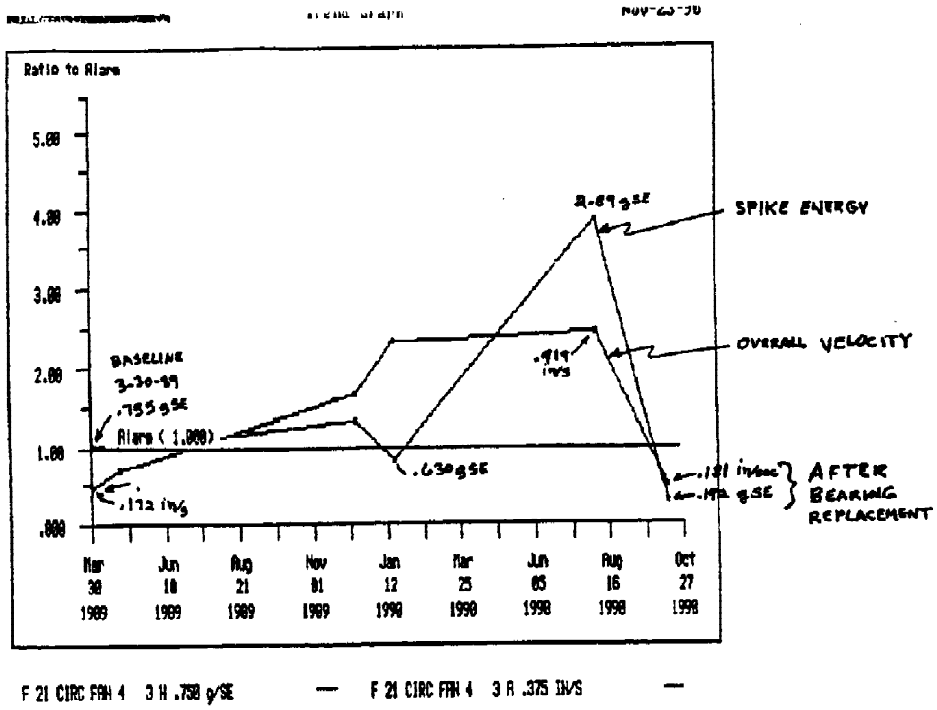

IRD Mechanalysis Data Analysis System

Machine F 21 CIRC FAN 4 Pos 3 Dir H Units g/SE Alarm .750

Ampl	Insp Code	Date mm/dd/yy	Time	Ampl	Insp Code	Date mm/dd/yy	Time
.755	0	03-30-89	10:57				
.744	0	04-25-89	14:51				
.979	0	12-12-89	11:49				
.630	0	01-18-90	11:33				
2.89	0	08-03-90	14:05				
.192	0	10-11-90	15:07				

IRD Mechanalysis Data Analysis System

Machine F 21 CIRC FAN 4 Pos 3 Dir A Units IN/S Alarm .375

Ampl	Insp Code	Date mm/dd/yy	Time	Ampl	Insp Code	Date mm/dd/yy	Time
.172	0	03-30-89	10:57				
.263	0	04-25-89	14:51				
.628	0	12-12-89	11:49				
.872	0	01-18-90	11:33				
.919	0	08-03-90	14:04				
.181	0	10-11-90	15:07				

FIGURE 3.

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IRD Mechanalysis Data Analysis System
Nov-23-90

Machine Report (Latest Data) for: F 21 CIRC FAN 4

Date	Time	Pos	Dir	Ampl	Units	Change	Alarm	Inspection
Oct-11-90	15:05	1	A	.285	IN/S	21%	.375	
Oct-11-90	15:05	1	H	*.536	IN/S	267%	.375	
Oct-11-90	15:05	1	H	.012	g/SE	-94%	.500	
Oct-11-90	15:05	1	V	*.466	IN/S	134%	.375	
Oct-11-90	15:05	2	A	.096	IN/S	-64%	.375	
Oct-11-90	15:06	2	H	.162	IN/S	31%	.375	10-11-90 MEASUREMENT REPORT
Oct-11-90	15:06	2	H	.028	g/SE	-81%	.500	
Oct-11-90	15:06	2	V	.275	IN/S	-6%	.375	
Oct-11-90	15:07	3	A	.181	IN/S	-80%	.375	
Oct-11-90	15:07	3	H	.039	IN/S	-91%	.375	
Oct-11-90	15:07	3	H	.192	g/SE	-93%	.500	
Oct-11-90	15:07	3	V	.077	IN/S	-80%	.375	

12 lines this report

Hit any key to continue

IRD Mechanalysis Data Analysis System Page 1
Nov-23-90

Machine Report (Latest Data) for: F 21 CIRC FAN 4

Date	Time	Pos	Dir	Ampl	Units	Change	Alarm	Inspection
Aug-03-90	14:24	1	A	.236	IN/S	36%	.375	
Aug-03-90	14:24	1	H	.146	IN/S	-51%	.375	
Aug-03-90	14:24	1	H	.207	g/SE	527%	.500	
Aug-03-90	14:24	1	V	.199	IN/S	2%	.375	
Aug-03-90	14:24	2	A	.266	IN/S	29%	.375	
Aug-03-90	14:24	2	H	.124	IN/S	33%	.375	
Aug-03-90	14:24	2	H	.149	g/SE	51%	.500	
Aug-03-90	14:25	2	V	.293	IN/S	17%	.375	8-3-90 MEASUREMENT REPORT
Aug-03-90	14:04	3	A	*.919	IN/S	5%	.375	
Aug-03-90	14:05	3	H	*.452	IN/S	12%	.375	
Aug-03-90	14:05	3	H	*2.89	g/SE	776%	.500	
Aug-03-90	14:05	3	V	*.383	IN/S	22%	.375	
Aug-03-90	14:06	4	A	*.390	IN/S	-16%	.375	
Aug-03-90	14:06	4	H	*.627	IN/S	44%	.375	
Aug-03-90	14:06	4	H	*2.94	g/SE	167%	.500	
Aug-03-90	14:06	4	V	*.625	IN/S	101%	.375	

16 lines this report

FIGURE 4

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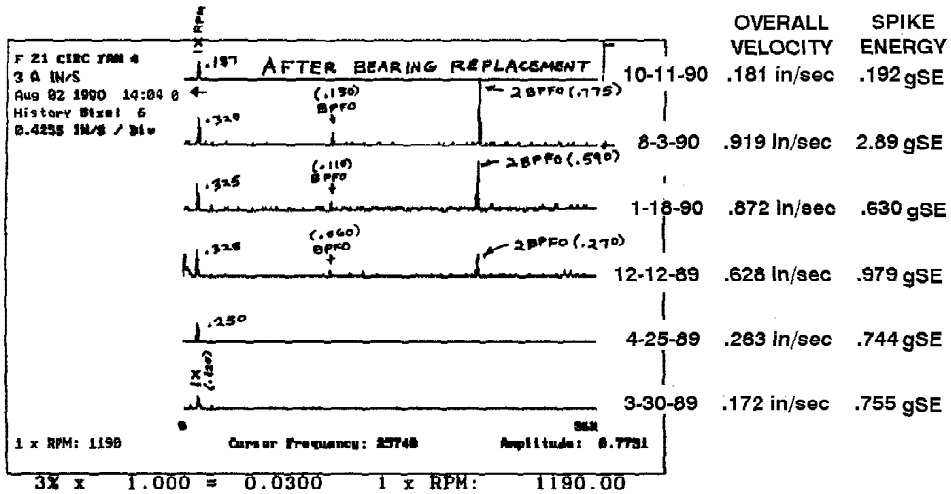

	OVERALL VELOCITY	SPIKE ENERGY
10-11-90	.181 in/sec	.192 gSE
8-3-90	.919 in/sec	2.89 gSE
1-18-90	.872 in/sec	.630 gSE
12-12-89	.628 in/sec	.979 gSE
4-25-89	.263 in/sec	.744 gSE
3-30-89	.172 in/sec	.755 gSE

3% x 1.000 = 0.0300 1 x RPM: 1190.00

Machine	Pos	Dir	Units	Date
F 21 CIRC FAN 4	3	A	IN/S	Aug 03 1990 14:04

Frequency	1 x RPM	Amplitude	
180	0.151	0.0450	
1170	0.983	0.3198	← 1X RPM
12870	10.815	0.1500	← BPFO
18090	15.202	0.0300	
23400	19.664	0.0349	-2X RPM
25740	21.630	0.7751	← 2BPFO
26910	22.613	0.0349	+1X RPM
28080	23.597	0.0450	+2X RPM
31500	26.471	0.0300	
32670	27.454	0.0450	
33840	28.437	0.0349	

004% x 1.000 = 0.0400 1 x RPM: 1190.00

Machine	Pos	Dir	Units	Date
F 21 CIRC FAN 4	3	A	IN/S	Jan 18 1990 11:33

Frequency	1 x RPM	Amplitude	
1170	0.983	0.3250	← 1X RPM
2340	1.966	0.0400	
12870	10.815	0.1099	← BPFO
25650	21.555	0.5900	← 2BPFO
26820	22.538	0.0850	← +1X RPM
31320	26.319	0.0400	
32580	27.378	0.0750	
33840	28.437	0.0650	
34380	28.891	0.0400	
34920	29.345	0.0800	
35190	29.571	0.0400	

003% x 1.000 = 0.0300 1 x RPM: 1190.00

Machine	Pos	Dir	Units	Date
F 21 CIRC FAN 4	3	A	IN/S	Dec 12 1989 11:49

Frequency	1 x RPM	Amplitude	
90	0.076	0.2650	
360	0.303	0.0950	
1170	0.983	0.3250	← 1X RPM
12870	10.815	0.0600	← BPFO
15750	13.235	0.0400	
25740	21.630	0.2701	← 2BPFO
26910	22.613	0.0300	
29070	24.429	0.0300	
30240	25.412	0.0300	
31500	26.471	0.0300	
33300	27.983	0.0400	
33840	28.437	0.0400	
34380	28.891	0.0300	

FIGURE 5

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SERIOUS PROBLEM OF A BEARING TURNING ON A SHAFT OF ONE MOTOR AND DETECTION OF SEVERE BEARING PROBLEMS ON ANOTHER MOTOR AS EVIDENCED BY NUMEROUS OUTER RACE SUM AND DIFFERENCE FREQUENCIES

I. BACKGROUND

On Saturday March 12, it is understood that the Rolling Mill #4 Main Drive Motor had high vibration and noise amplitudes which could even be felt in the floor and heard in other areas. Technical Associates was contacted by the plant on March 15 and brought in to diagnose the problem. By this time, the noise and vibration had dropped significantly, but there was a peculiar noise pulse present each time the machine was started up under load, particularly at lower speeds up to approximately 300 to 400 RPM. Measurements were made on the #4 Main Drive Motor using a real-time analyzer. In addition, an analysis was conducted on the #4 Rewind Inboard Motor which was experiencing high vibration. Following is a discussion of the measurements that were made on March 15. Measurements were made with a Rockland 5815A narrowband real-time analyzer.

Later, after corrective actions were completed, a Predictive Maintenance Program was instituted at the client's plant by Technical Associates. This allowed capture of vibration spectra following the corrective actions using an IRD 890 and IRD 7090 software.

II. CONCLUSIONS

A. MILL 4 MAIN DRIVE MOTOR (See Figure 1)

Vibration spectra indicate the following:

1. No vibration spectra on this machine exceeded .300 in/sec alarm levels on this March 15 test date at any of the six (6) positions evaluated on the outboard and inboard motor bearings (See Plots 2 and 3 containing spectra for all 6 positions tested).

2. Bearing Health - No bearing frequencies which would indicate defects were found. Therefore, this indicated motor bearings were likely in good condition.

3. Balance and Alignment - All spectra indicated a well balanced rotor assembly with good alignment.

4. Electrical Condition - No electrical problems were evident from the spectra based on signatures taken in the morning with a lighter gauge metal product, nor were any evident in the afternoon when a heavier gauge, non-annealed product was run which put the rolling mill drive under heavier strain.

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5. Mechanical Looseness - Definite mechanical looseness was indicated in spectra taken at all 6 motor positions, based on the presence of a series of motor speed harmonics (multiples of RPM). On the March 15 test day, it was verbally reported that the source of looseness was likely either between the inboard bearing surface and shaft journal or at the female bearing seat supporting the housing. Since highest overall levels on March 15 were all below .20 in/sec (compared to .30 in/sec Alarm level), there was no indication of imminent failure at that time. However, because plant maintenance engineering reported much higher amplitudes had been present on March 12, and due to the presence of pronounced pulsing noises each time the motor was brought up to speed, concern was expressed on the health of the machine despite the fact that vibration was now, on March 15, below alarm.

B. MILL 4 REWIND MOTOR (See Figure 2)

1. In general, vibration was about 5 times higher on March 15, 1988 than it was when similar spectra were captured on the #4 Inboard Rewind Motor on December 10, 1987. All spectra were dominated by the presence of bearing sum and difference frequencies indicating defects on the outer race. Vibration spectra indicated motor bearings in advanced state of wear. On March 15, it was recommended that both motor bearings be replaced ASAP.

2. The motor still seemed well balanced. There was no evidence of electrical problems at that time. However, slight misalignment was indicated between the outboard and inboard motors and/or there seemed to be a slight problem with the electric clutch assembly located between the two motors (see Figure 2).

III. RECOMMENDATIONS (Made before failure of the Mill 4 Main Drive Motor)

A. MILL 4 MAIN DRIVE MOTOR

Because of the definite mechanical looseness symptoms described in “Conclusions” Section II-A5, it was recommended that the plant take readings each day to monitor possible deterioration of the problem, and also to see if any defect frequencies for the motor bearings would appear. It was not felt that any significant damage had yet been done to the bearings due to the complete absence of defect frequencies (this could be asserted with particular confidence since defect frequencies for this particular SKF 6236 bearing were well removed from integer running speed harmonics - BPFI = 6.338X; BPFO = 4.662X; BSF = 3.205X). In fact, a series of zoom spectra confirmed that no noticeable frequencies other than running speed harmonics and gear mesh frequencies were present. It was further recommended that measurement intervals could be increased to weekly and later monthly if no further damage was evident in future vibration spectra.

UPDATE:

Later, it was reported that the Mill 4 Main Drive Motor did experience failure on March 18. Inspection confirmed the mechanical looseness, and found the source at the inboard motor bearing, revealing that the shaft journal had been turning in the bearing removing .126" from the shaft diameter. In all probability, much of this damage was done on March 12 when all the high vibration and noise was noticed by the plant. The inspection also confirmed that the bearing itself was still in fact in good shape as had been indicated by the vibration signatures, despite the excessive damage to the shaft.

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B. MILL 4 REWIND MOTOR

It was strongly recommended that the plant take immediate steps to replace the motor bearings due to both the amplitude and the presence of a great number of the difference frequencies spaced at the SKF 6218 outer race defect frequency (4.11X RPM). In addition, it was recommended that the plant carefully check the alignment of the rewind drive assembly as well as the condition of the electric clutch assembly located between the outboard and inboard motors (see Figure 2).

IV. DISCUSSION

A. MILL 4 MAIN DRIVE MOTOR

A series of axial, horizontal and vertical measurements was made on the main drive motor bearings in order to assess the condition of the motor (see Figure 1). The vibration spectra derived from these measurements indicated the following:

1. No vibration levels exceeding alarm were present on either motor bearing on this test day.

2. No bearing defect frequencies were present, indicating good bearing condition.

3. Balance and alignment indications were good.

4. No motor electrical problems were present, even when the mill was operating with heavy gage material.

5. The dominant feature of all the spectra taken was the large number of harmonics of running speed, an indication of mechanical looseness in the motor. A verbal diagnosis at the plant was made that the most likely problem was looseness of the inboard bearing seat in which the bearing housing was held (closest to gearbox) and/or looseness between the bearing surface and shaft journal.

A visual inspection of the inboard bearing was made with no concrete conclusions, but there was some indication that the bearing seat could be loose.

Later, it is understood that a shutdown was mandated on March 18 because of sparks and other signs of imminent failure. Disassembly and inspection by the plant revealed that on the inboard end of the motor, the shaft journal had been turning in the bearing and the journal had been eroded to the point of being .126 inch undersized. The journal, normally a press fit on the bearing, now had a very large clearance, leaving the motor shaft loose on the inboard end. This explains the mechanical looseness indicated by the March 15 measurements, an example of which is shown by Plot 1, derived from a measurement at the inboard bearing of the motor. Note the large number of harmonics of running speed present in the spectrum, a typical spectrum for mechanical looseness. For comparison purposes, Plots 2 and 3 are included showing spectra taken at all 6 motor positions located on the outboard and inboard bearing housings. Note that mechanical looseness is evident in all 6 spectra, based on the large number of 567 RPM running speed harmonics. However, note that all overall levels remained below the alarm level for this machine, which would be about **.300 in/sec** (overalls ranged from .074 in/sec to .176 in/sec).

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The motor was disassembled and the shaft was rebuilt by General Electric Service Shop. It is understood that the motor has been put back into service and is apparently operating satisfactorily. It is recommended that signatures be obtained to check current machine health and to serve as **Baseline Signatures** for future reference.

B. MILL 4 REWIND MOTOR

Measurements made on the Mill 4 Inboard Rewind Motor indicated definite bearing problems with the SKF 6218 outboard bearing. While measurements were being made, a large number of difference frequencies were noticed. The spacing of these difference frequencies was equal to the outer race defect frequency (4.11 F). The upper spectrum of Plot 4 shows a high overall of **.544 in/sec**, dominated by the outer race difference frequencies. The lower signature is a zoom spectrum showing that the difference frequencies were spaced at 2280 CPM when the motor speed was 555 RPM - equal to 4.11 times the motor speed confirming the presence of bearing wear. For comparison purposes, spectra taken at all 6 positions are provided in Plots 5 and 6. They likewise showed wear of the outboard bearing due to high outer race frequency difference frequencies.

Therefore, on the March 15 test day, it was recommended that both of the bearings be changed in the inboard rewind motor, a recommendation which has been carried out. The motor is now back in service and is apparently functioning well.

V. CORRECTIVE ACTIONS TAKEN

A. MILL 4 MAIN DRIVE MOTOR

Upon investigation, the plant found that one of the problems which may have contributed to the bearing failure was too light a press fit. This bearing had a bore of 180 mm (7.086 in), but a press fit of only .0009 inch. It was decided to increase this to a M6 fit which ranges from .0006 to .0030 inch in the new motor bearings. In addition, an investigation also found that when this variable speed machine ran at certain speeds, the gear mesh frequency transmitting from the gearbox into the motor was exciting an axial natural frequency of the motor. Thus, it was decided not to operate the motor in this particular speed range in which it was resonant.

B. MILL 4 REWIND MOTOR

Each of the motor bearings were first replaced. Then, a slight misalignment problem was corrected.

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VI. RESULTS OF CORRECTIVE ACTIONS

Vibration spectra were captured with an IRD 890 analyzer after corrective actions were completed as part of a plant predictive maintenance program.

A. MILL 4 MAIN DRIVE MOTOR

Plots 7A and 7B compare spectra taken before and after motor bearing replacement respectively (note that the PMP route spectra had an F_{MAX} of 60,000 CPM compared to 12,000 CPM on the Plot 7A spectrum taken during the earlier diagnostic investigation). Importantly, note that no harmonics at running speed were present in the "After" spectrum of Plot 7B (at least none were present exceeding the .004 in/sec threshold value). Note also that the overall velocity dropped from .176 in/sec to .059 in/sec.

B. MILL 4 REWIND MOTOR

Plots 8A and 8B compare "Before" and "After" spectra on the Mill 4 Rewind Motor. Note the dramatic drop in overall velocity from .544 in/sec to only .018 in/sec - a reduction of over 96%. Also, note the tremendous drop in higher frequency vibration, particularly between 20,000 and 60,000 CPM.

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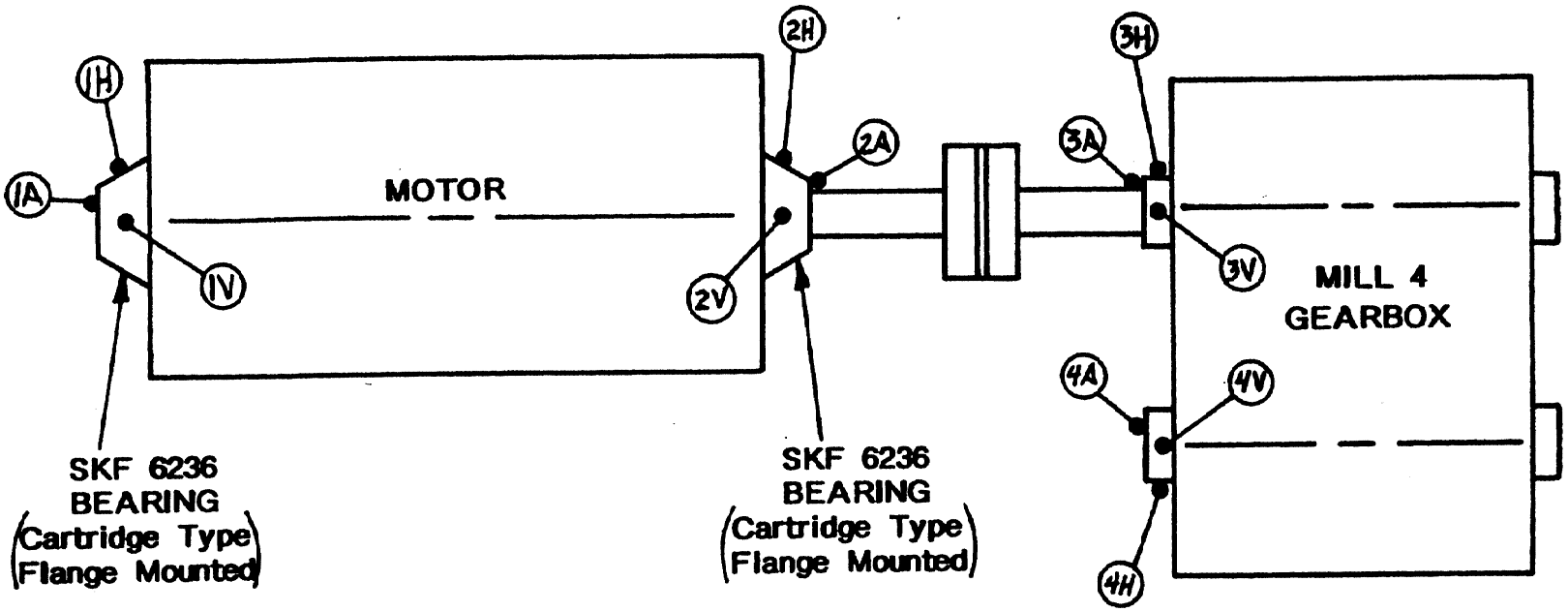
FIGURE 1
MILL 4 MAIN DRIVE ARRANGEMENT



SKF 6236 BEARING DEFECT FREQUENCIES AT 567 RPM:

BPFI = 6.338 X RPM = 6.338 X 567 = 3594 CPM
BPFO = 4.662 X RPM = 4.662 X 567 = 2643 CPM
BSF = 3.205 X RPM = 3.205 X 567 = 1817 CPM
FTF = 0.424 X RPM = 0.424 X 567 = 240 CPM

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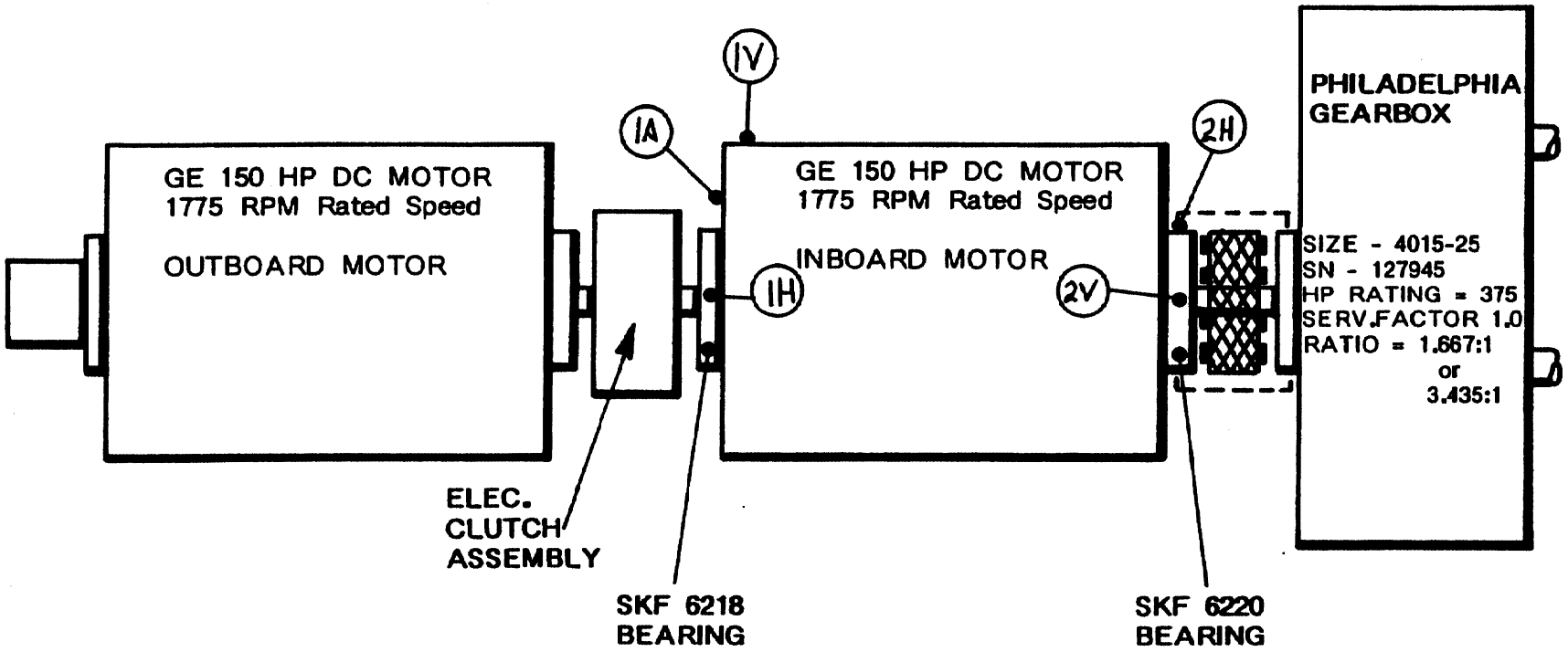
FIGURE 2
MILL 4 REWIND DRIVE ASSEMBLY



MILL 4 REWIND MOTOR BEARING DEFECT FREQUENCIES AT 555 RPM

BEARING	BPFI	BPFO	BSF	FTF
SKF 6220	5,907 X = 3278 CPM	4.093 X = 2272 CPM	2.665 X = 1479 CPM	0.409 X = 227 CPM
SKF 6218	5,889 X = 3268 CPM	4.111 X = 2282 CPM	2.723 X = 1511 CPM	0.411 X = 228 CPM

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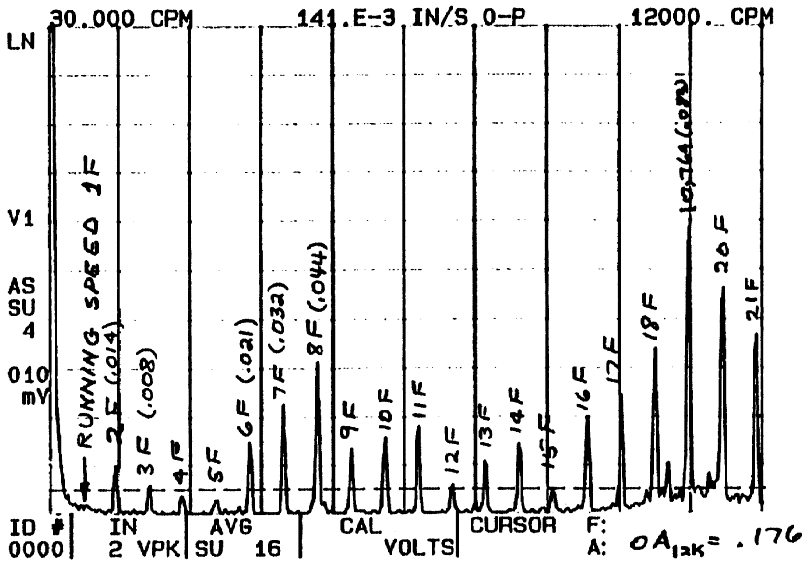
PLOT 1



PEAK PICK ON TRACE 1

RPM = 567

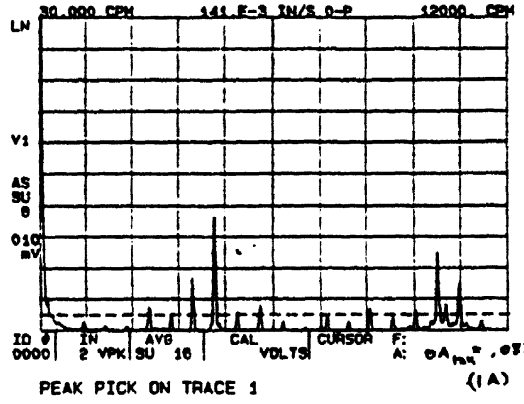
POINT 2A ON MOTOR

ALL SIGNIFICANT SPIKES ARE HARMONICS OF RUNNING SPEED

MECHANICAL LOOSENESS IS INDICATED

IN/S 0-P	CPM
13.5E-3	1140.0 = 2F
8.37E-3	1710.0 = 3F
21.2E-3	3390.0 = 6F
32.2E-3	3959.9 = 7F
44.4E-3	4530.0 = 8F
19.3E-3	5100.0 = 9F
22.5E-3	5670.0 = 10F
25.8E-3	6240.0 = 11F
8.37E-3	6810.0 = 12F
15.5E-3	7350.0 = 13F
20.6E-3	7920.0 = 14F
7.08E-3	8490.0 = 15F
28.3E-3	9060.0 = 16F
34.8E-3	9630.0 = 17F
48.3E-3	10200. = 18F
7.08E-3	10320.
14.8E-3	10410.
83.0E-3	10769. = 19F
7.73E-3	10919.
11.6E-3	11100.
9.01E-3	11159.

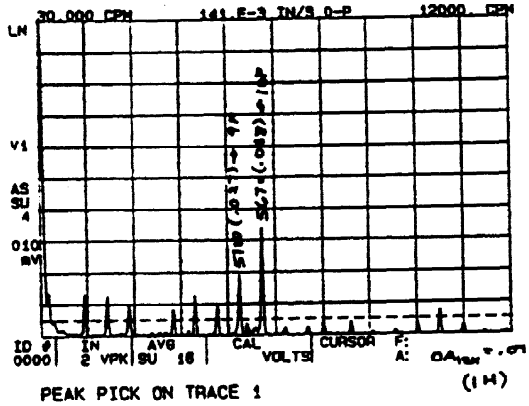
MILL 4 MAIN DRIV MOTOR



Position 1A
(Outboard motor Brg)

IN/S 0-P	CPM
14.2E-3	210.00
7.08E-3	330.00
10.3E-3	2820.0
23.8E-3	3959.9
51.5E-3	4530.0
8.37E-3	5100.0
10.9E-3	5670.0
9.66E-3	8490.0
9.01E-3	9630.0
35.4E-3	10200.
11.6E-3	10410.
21.2E-3	10769.

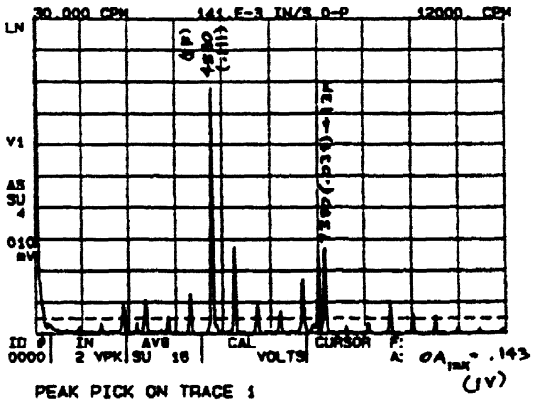
MIL4 MDMOT



Position 1H

IN/S 0-P	CPM
19.3E-3	240.00
7.08E-3	360.00
18.7E-3	1140.0
17.4E-3	1710.0
13.5E-3	2250.0
11.6E-3	3390.0
18.0E-3	3959.9
14.2E-3	4530.0
27.0E-3	5100.0
47.6E-3	5670.0
10.3E-3	10200.

MIL4 MDMOT



Position 1V

IN/S 0-P	CPM
13.5E-3	2250.0
15.5E-3	2820.0
7.73E-3	3390.0
18.0E-3	3959.9
111.E-3	4530.0
39.3E-3	5100.0
13.5E-3	5670.0
10.3E-3	6240.0
24.5E-3	6810.0
34.1E-3	7200.0
38.6E-3	7379.7
13.5E-3	9060.0
9.01E-3	9630.0
8.37E-3	10200.

MIL4 MDMOT

PLOT 2
MILL 4 MAIN DRIVE MOTOR POSITIONS 1A, 1H, & 1V

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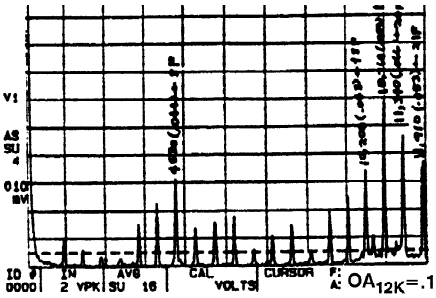
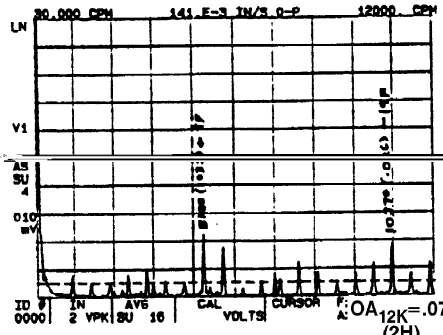


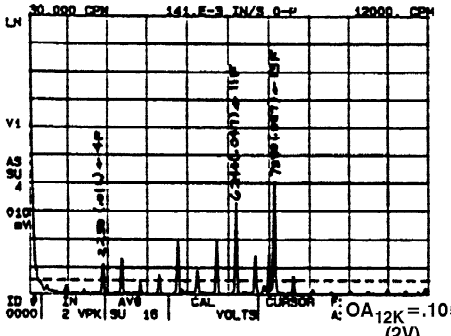
IN/S 0-P	CPM
13.5E-3	1140.0
8.37E-3	1710.0
21.2E-3	3390.0
32.2E-3	3959.9
44.4E-3	4530.0
19.3E-3	5100.0
22.5E-3	5670.0
25.8E-3	6240.0
8.37E-3	6810.0
15.5E-3	7350.0
20.6E-3	7920.0
7.08E-3	8490.0
28.3E-3	9060.0
34.8E-3	9630.0
48.3E-3	10200.
7.08E-3	10320.
14.8E-3	10410.
83.0E-3	10769.
7.73E-3	10919.
11.6E-3	11100.
9.01E-3	11159.

MILL 4 MAIN DRIVE MOTOR



IN/S 0-P	CPM
12.2E-3	240.00
10.3E-3	300.00
7.08E-3	360.00
10.9E-3	2820.0
12.9E-3	3390.0
7.08E-3	3600.0
7.73E-3	3959.9
7.08E-3	4530.0
31.5E-3	5100.0
25.1E-3	5670.0
7.73E-3	6810.0
10.9E-3	7200.0
9.01E-3	7350.0
17.4E-3	7920.0
12.2E-3	8490.0
7.08E-3	9060.0
16.7E-3	10200.
26.4E-3	10769.
10.9E-3	11340.

MILL 4 MD MOTOR



IN/S 0-P	CPM
14.2E-3	240.00
15.5E-3	2250.0
18.7E-3	2820.0
10.3E-3	3959.9
28.3E-3	4530.0
12.2E-3	5100.0
27.7E-3	5670.0
47.0E-3	6240.0
20.0E-3	6810.0
20.6E-3	7200.0
57.3E-3	7379.7
9.66E-3	7920.0

PLOT 3

MILL 4 MAIN DRIVE MOTOR POSITIONS 2A, 2H, & 2V

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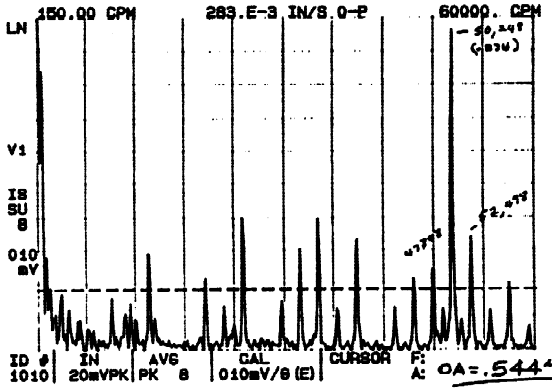
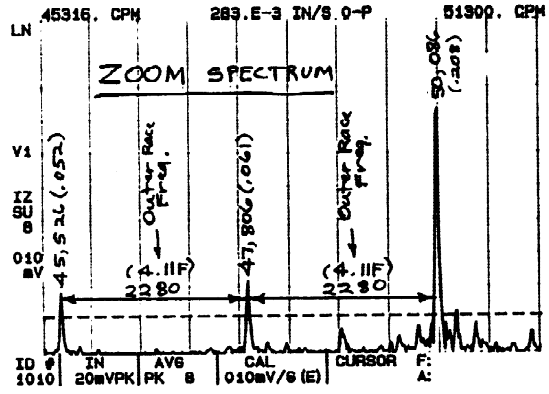


IN/S 0-P	CPM
239.E-3	600.00
80.1E-3	1200.0
53.0E-3	1649.9
82.7E-3	13649.
60.7E-3	20550.
114.E-3	25049.
86.6E-3	31950.
112.E-3	34200.
94.3E-3	38850.
60.7E-3	45600.
68.5E-3	47848.
274.E-3	50248.
96.9E-3	52498.
56.9E-3	57000.

Position 1A
Outboard Bearing
(Inboard Motor)
MOTOR SPEED = 555 RPM
(at this instant)

MIL4 REWMT



IN/S 0-P	CPM
51.7E-3	45526.
60.7E-3	47806.
208.E-3	50086.
34.9E-3	50310.

Position 1A
Current Motor Speed = 555 RPM (at this instant)
2280 CPM = $\frac{2280}{555}$ = (4.11F) = OUTER RACE BEARING FREQUENCY

MIL4 REWMT

PLOT 4
MILL 4 REWIND MOTOR POSITION 1A

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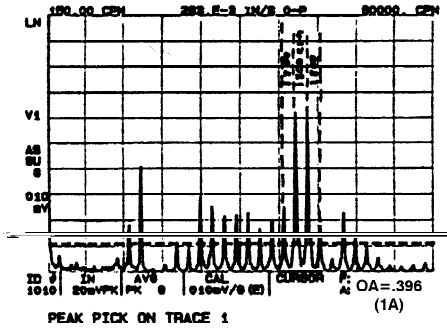
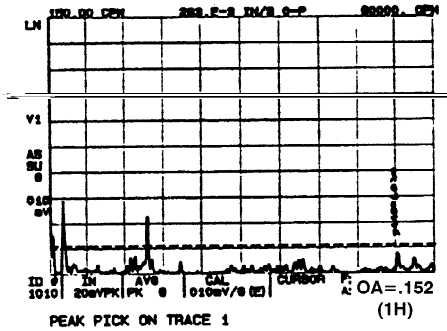


Position 1A
Motor Speed ≈ 450 RPM
(Towards End of Cycle)
(Sidebands ≈ 1800 CPM)
Mill 4 Rewind Motor

IN/S 0-P	CPM
51.7E-3	12899.
115.E-3	14700.
81.4E-3	23849.
71.1E-3	25649.
60.7E-3	27600.
62.0E-3	29399.
64.6E-3	31199.
46.5E-3	33000.
56.9E-3	34950.
69.8E-3	36750.
176.E-3	38550.
36.2E-3	39000.
182.E-3	40350.
40.1E-3	40800.
50.4E-3	42150.
64.6E-3	45898.
38.8E-3	47700.

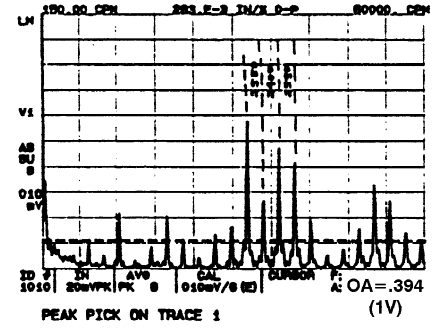
MIL4 REWMT



Position 1H

IN/S 0-P	CPM
42.6E-3	600.00
63.3E-3	15449.

MIL4 REWMT



Position 1V

IN/S 0-P	CPM
96.9E-3	600.00
32.3E-3	1200.0
60.7E-3	12450.
58.1E-3	19800.
37.5E-3	27300.
45.2E-3	29699.
31.0E-3	31650.
164.E-3	32249.
74.9E-3	34800.
133.E-3	37200.
118.E-3	39750.
53.0E-3	42150.
43.9E-3	49650.
93.0E-3	52050.
74.9E-3	54598.
38.8E-3	57000.
28.2E-3	7500
23.8E-3	17400

PLOT 5 MIL4 REWMT

MILL 4 REWIND MOTOR (POSITIONS 1A, 1H, & 1V)

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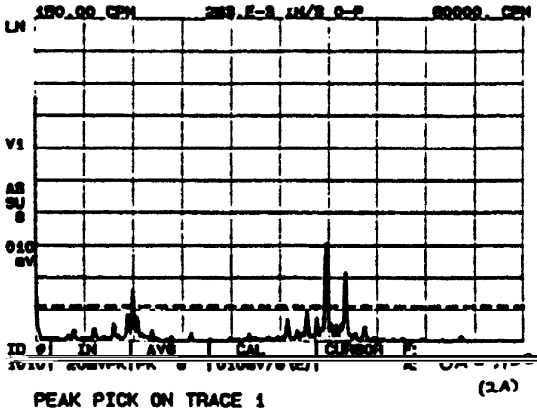
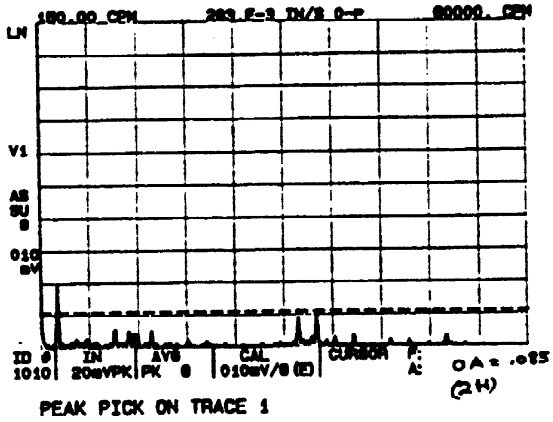


IN/S 0-P	CPM
45.2E-3	11850.
86.6E-3	35700.
60.7E-3	38100.
19.9E-3	30900.
20.5E-3	33300

Position 2A

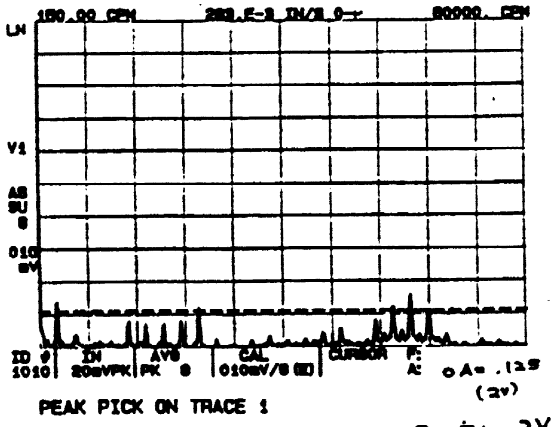
MIL4 REWMT



IN/S 0-P	CPM
55.6E-3	2250.0
25.4E-3	31800
27.9E-3	34050

Position 2H

MIL4 REWMT



IN/S 0-P	CPM
38.8E-3	2250.0
33.6E-3	19650.
33.6E-3	43498.
43.9E-3	45750.
31.0E-3	47848.
22.4E-3	10950
19.4E-3	13050
19.3E-3	15300
22.4E-3	17400
34.1E-3	19650
23.2E-3	41400

Position 2V

PLOT 6
MILL 4 REWIND MOTOR (POSITIONS 2A, 2H, & 2V)

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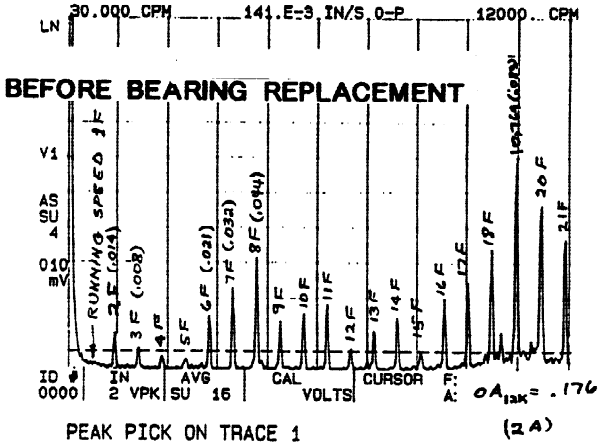


(REPEAT OF PLOT 1)
RPM = 567

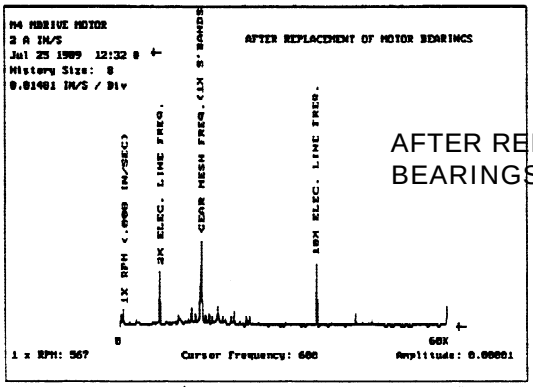
POINT 2A ON MOTOR
ALL SIGNIFICANT SPIKES ARE HARMONICS OF RUNNING SPEED

MECHANICAL LOOSENESS IS INDICATED

IN/S 0-P	CPM
13.5E-3	1140.0 = 2F
8.37E-3	1710.0 = 3F
21.2E-3	3390.0 = 6F
32.2E-3	3959.9 = 7F
44.4E-3	4530.0 = 8F
19.3E-3	5100.0 = 9F
22.5E-3	5670.0 = 10F
25.8E-3	6240.0 = 11F
8.37E-3	6810.0 = 12F
15.5E-3	7350.0 = 13F
20.6E-3	7920.0 = 14F
7.08E-3	8490.0 = 15F
28.3E-3	9060.0 = 16F
34.8E-3	9630.0 = 17F
48.3E-3	10200. = 18F
7.08E-3	10320.
14.8E-3	10410.
83.0E-3	10769. = 19F
7.73E-3	10919.
11.6E-3	11100.
9.01E-3	11159.

MILL 4 MAIN DRIVE MOTOR

PLOT 7A



AFTER REPLACEMENT OF MOTOR BEARINGS OVERALL = .059 IN/SEC

002% x 0.2000 = 0.00400 1 x RPM: 567.00

Machine		POS	DIR	Units		Date
M4 MDRIVE MOTOR		2	A	IN/S		Jul 25 1989 12:32
Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude	
150	0.26	0.00550	36000	63.49	0.03050	
600	1.06	0.00801	43200	76.19	0.00648	
7200	12.70	0.02701	59850	105.56	0.01048	
10500	18.52	0.00600				
10950	19.31	0.00400				
12900	22.75	0.00950				
13500	23.81	0.00648				
14700	25.93	0.04149				
15300	26.98	0.00550				
15900	28.04	0.00648				
16500	29.10	0.00448				
17550	30.95	0.01048				
18450	32.54	0.00448				
19950	35.19	0.00448				
20550	36.24	0.00750				
22800	40.21	0.00499				
23400	41.27	0.00499				

T - Threshold adjust Q - Quit

PLOT 7B

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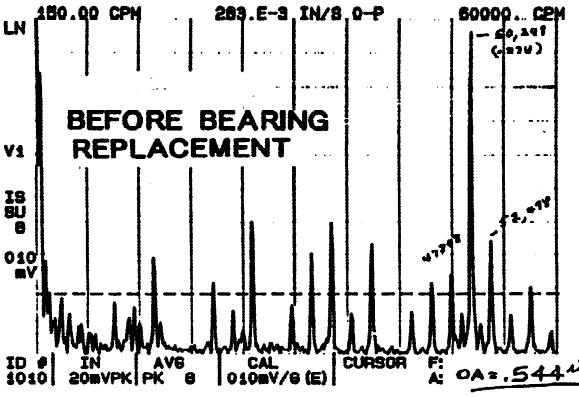


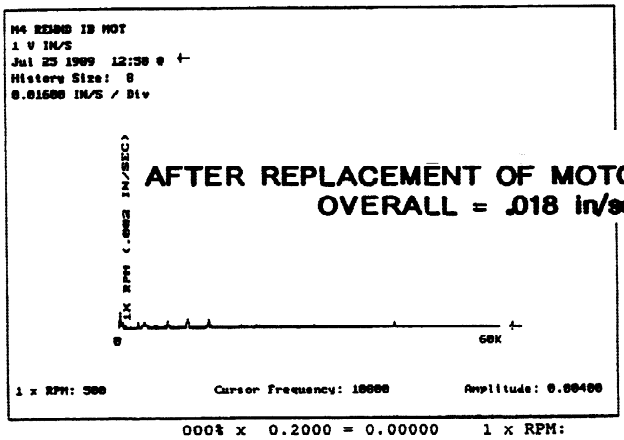
PEAK PICK ON TRACE 1

(REPEAT OF PLOT 4) Position 1A Outboard Bearing (Inboard Motor)

MOTOR SPEED = 555 RPM (at this instant)

IN/S 0-P	CPM
239.E-3	600.00
80.1E-3	1200.0
53.0E-3	1649.9
82.7E-3	13649.
60.7E-3	20550.
114.E-3	25049.
86.6E-3	31950.
112.E-3	34200.
94.3E-3	38850.
60.7E-3	45600.
68.5E-3	47848.
274.E-3	50248.
96.9E-3	52498.
56.9E-3	57000.

PLOT 8A



000% x 0.2000 = 0.00000 1 x RPM: 500.00

Machine	Pos	Dir	Units	Date
M4 REWND IB MOT	1	V	IN/S	Jul 25 1989 12:50

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
150	0.30	0.00699	43200	86.40	0.00248
600	1.20	0.00200			
900	1.80	0.00098			
3000	6.00	0.00200			
4050	8.10	0.00248			
5550	11.10	0.00098			
5850	11.70	0.00098			
7200	14.40	0.00149			
7650	15.30	0.00349			
10800	21.60	0.00400			
13950	27.90	0.00400			
14400	28.80	0.00098			
19200	38.40	0.00098			
21600	43.20	0.00098			
22050	44.10	0.00098			
30000	60.00	0.00098			
30450	60.90	0.00149			

T - Threshold adjust Q - Quit

PLOT 8B

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DETECTION OF A BALL BEARING PROBLEM ON A GLAND STEAM EXHAUST FAN WHOSE BEARING MANUFACTURER AND MODEL NUMBER WERE UNKNOWN

I. BACKGROUND

A utility company contracted Technical Associates to provide them a complete vibration analysis predictive maintenance program. This program began by acquisition of baseline measurements on all machines selected followed by data analysis and composition of a **Baseline Report**. For the most part, little was known about the machines except for nameplate data. One of the machines tested was the Gland Steam Exhaust Fan. This unit was an integrally mounted fan (shaft mounted) with anti-friction bearings and a nominal running speed of 3500 RPM. The machine arrangement is shown in Figure 1.

II. DISCUSSION OF RESULTS

IRD Mechanalysis Data Analysis System Page 1
Dec-04-89

TABLE I.

Machine Report (Latest Data) for: GLND STM EXH 61

Date	Time	Pos	Dir	Ampl	Units	Change	Alarm	Inspection
Aug-24-89	15:00	1	A	.286	IN/S	-----%	.375	
Aug-24-89	15:00	1	H	.171	IN/S	-----%	.375	
Aug-24-89	15:00	1	H	1.14	g/SE	-----%	1.25	
Aug-24-89	15:00	1	O	.176	IN/S	-----%	.375	
Aug-24-89	15:00	1	V	.211	IN/S	-----%	.375	
Aug-24-89	15:02	2	H	.208	IN/S	-----%	.375	
Aug-24-89	15:02	2	H	*1.32	g/SE	-----%	1.25	
Aug-24-89	15:02	2	V	*.401	IN/S	-----%	.375	

8 lines this report

Table I shows the list of overall levels of vibration (in/sec) and spike energy (gSE). Note from Table I that both spike energy (gSE) measurements on the outboard and inboard bearings were relatively high (1.14 gSE on the outboard Position 1 bearing and 1.32 gSE on the inboard bearing Position 2). The Position 2H spike energy exceeded the alarm as did the Position 2 vertical (2V) overall velocity measurement (.401 in/sec compared to an alarm of .375 in/sec). The Position 1V spectrum (Figure 2) had a well defined frequency at 76,200 CPM with running speed (3600 CPM nominal) sidebands. The Position 2V spectrum shown in Figure 3 had a moderately large 1X RPM amplitude of .350 in/sec and the same 76,200 CPM frequency, again with running speed sidebands.

III. CONCLUSIONS

A. The 76,200 CPM frequency seen on both motor bearings was likely the natural frequency of one of the bearing components (such bearing component natural frequencies are commonly found between approximately 30,000 and 120,000 CPM). When such bearing natural frequencies are sufficiently excited to cause them to appear in the velocity spectrum, this normally indicates the bearing is passing through the second stage of wear (such rolling element bearings usually have four rather well defined wear stages until eventual failure). Normally, when 1X RPM sidebands then appear around the bearing natural frequency, this signals a more serious problem and that the bearing is nearing the end of Stage 2. At this point, the bearing usually has only about 5% remaining life.

B. The fan wheel is unbalanced.

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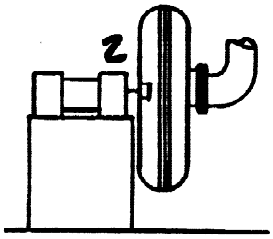
MACHINERY DATA GUIDE SHEET

RELIABILITY SURVEY | predictive maintenance program | (Page__of__)

UNITS
51
52
(61)
62

DETECTION OF A BALL BEARING PROBLEM ON A GLAND STEAM EXHAUST FAN WHOSE BEARING MANUFACTURER AND MODEL NUMBER WERE UNKNOWN



MACHINE IDENT: GLAND STEAM EXHAUST FAN

AREA:_____

ROUTE:_____

DRIVER	INTERMEDIATE	DRIVEN
RPM min : 3500	RPM in :	RPM min : 3500
max :	out :	max :
MFGR :	RATIO:	MFGR. :
MODEL :	MODEL :	MODEL :
SER NO. :	SER NO. :	SER NO. :
V___ P___ A___	RATING :	RATING :
SHEAVE :___DIA.	BELT # :	#BLADES :
BRG#: A (I)	BRG#:___(I)	BRG#: A (I)
BRG#:___(O)	BRG#___(O)	BRG#:___(O)
H.P.: 10	GEAR T:___CT.	GEAR T:___CT.
OTHER:	OTHER:	OTHER:

IF - 08

FIGURE 1

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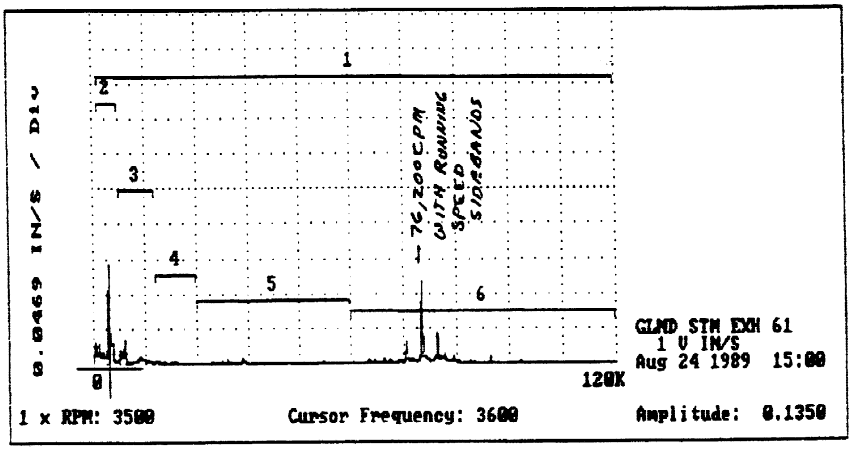

002% x 0.500 = 0.0100 1 x RPM: 3500.00

Machine	Pos	Dir	Units	Date
GLND STM EXH 61	1	U	IN/S	Aug 24 1989 15:00

Frequency	1 x RPM	Amplitude
300	0.09	0.0300
1200	0.34	0.0250
2100	0.60	0.0150
3600	1.03	0.1350
6000	1.71	0.0175
7200	2.06	0.0325
10800	3.09	0.0100
72600	20.74	0.0300
76200	21.77	0.1100
79500	22.71	0.0400
81300	23.23	0.0125
83400	23.83	0.0100

T - Threshold adjust Q - Quit

PLOT 27

FIGURE 2

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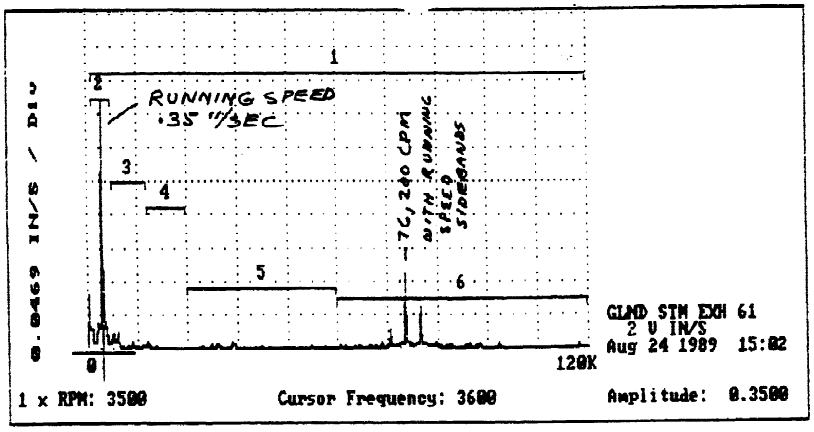

002% x 0.500 = 0.0100 1 x RPM: 3500.00

Machine	Pos	Dir	Units	Date
GLND STM EXH 61	2	V	IN/S	Aug 24 1989 15:02

Frequency	1 x RPM	Amplitude	
300	0.09	0.0775	
2400	0.69	0.0350	
3600	1.03	0.3500	
6300	1.80	0.0200	
7200	2.06	0.0225	
14400	4.11	0.0100	
72600	20.74	0.0250	— 1x RPM S.B.
76200	21.77	0.1050	←
79800	22.80	0.0575	— 1x
83400	23.83	0.0100	— 2x

T - Threshold adjust Q - Quit

PLOT 28

FIGURE 3

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III. RECOMMENDATIONS

A. Since the signatures on both bearings indicated late Stage 2 wear, and because spike energy levels were likewise high on each, it was recommended that both motor bearings be replaced.

B. It was then recommended the plant reevaluate the unbalance condition after the bearings were replaced. If both signature and phase analysis then still indicated unbalance, the plant was instructed to clean and balance the fan wheel (see Figure 4).

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SUMMARY OF RESULTS AND RECOMMENDATIONS FOR MACHINES FOUND TO HAVE PROBLEMS
(SORTED FIRST BY PROBLEM PRIORITY, THEN BY PROBLEM SEVERITY WITHIN PRIORITY)

PAGE 2 of 7 — AS OF: 8/24/89

AREA	MACHINE COMPUTER NAME	INDICATED PROBLEM(S): UNBALANCE	MISALIGNMENT/SOFT FOOT	BENT SHAFT	LUBRICATION PROBLEM	BEARING PROBLEM	MECHAN. LOOSENESS	BELT/SHEAVE PROBLEM	RESONANCE PROBLEM	GEAR PROBLEM	BLADE PASSING PROB.	CAVITATION	MOTOR ELECTRICAL	OIL WHIRL/WHIP	LOOSE FOUNDATION	VIB. MONITOR PROB.	SEAL/ROTOR RUB	OTHER (SEE REMARKS)	PRIORITY OF PROBLEM	HIGHEST OVERALL VIBRATION: POSITION NO.	HIGHEST OVERALL VIBRATION: LEVEL (9 = mils, 3 = in/sec, 9's)	RECOMMENDATIONS & REMARKS	SEE PLOT NO.
	GLND STM EXH 61	●				●													①	1V 2V	.211 .401	Band 6 alarms at Positions 1V and 2V due to bearing defect frequency with running speed (3600 CPM) sidebands. Plot 28 also exhibits unbalance. Replace motor bearings then re-evaluate balance condition. Clean and balance if still required.	27 28
	TURB DRV BFP 51	●	●																2	1A 1V 2H 3A M4 V	.279 .510 .320 .309 2.30 MILS	Moderate unbalance and misalignment noted at Positions 1 and 2 on the pump and misalignment at Position 3 on the turbine. Suggest checking the alignment between the pump and turbine at Positions 2 and 3 during the next scheduled overhaul. Make any necessary corrections to the alignment. Also, inspect the impeller for foreign debris, removing same. If found, then re-evaluate unbalance and balance if still required. The non-contact monitor displacement (MILS) reading at Position M4 V at the turbine indicates a slight mechanical looseness. Trend all above conditions.	54 55 56 57 58.
	SERVWTR VPMP 61	●					●												2	1S-H	.478	Noticeable one times RPM vibration with harmonics indicating unbalance and mechanical looseness. Mechanical looseness may be from a worn bearing or loosened fasteners. Unbalance could be from foreign material caught in one of the impeller blades. Inspect for above and repair/replace as necessary.	46
	TURB DRV BFP 61	●																	2	1H 2H	.370 .458	Moderate unbalance noted at Positions 1 and 2 on the pump. During the next scheduled outage, suggest inspecting the impeller for foreign debris, removing same, if found. Then re-evaluate unbalance condition and balance if still necessary. A single measurement on the speed increaser showed a relatively high gear mesh amplitude but there is very little sideband indicating that the gears are probably OK. The Position 5V measurement shows a gear mesh amplitude of .290 in/sec. Plot 61. This should be carefully trended to insure that no upward trend occurs.	59 60 61

①= TAKE IMMEDIATE REPAIR ACTION DUE TO SEVERE PROBLEM(S) NOTED (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
2 = SCHEDULE REPAIRS TO CORRECT PROBLEMS NOTED AT CONVENIENCE (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
3 = DESIGNATED PROBLEM DOES NOT WARRANT ACTION AT THIS TIME. (CONTINUE ONLY WITH RELIABILITY SURVEY TREND ANALYSIS)

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(704) 333-9011

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FIGURE 4

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Electrically Induced High Frequencies: Their Resulting Effect on Bearings and What To Do About It

By Robert Hoskins
R. R. Donnelley & Sons Company
Gallatin, Tennessee

Introduction

After experiencing what we believed to be too frequent of motor bearing failures in the solvent recovery system and presses at the R.R. Donnelley printing plant in Gallatin, Tennessee, we decided to disassemble a motor ourselves and examine its condition. Primary on our minds was finding the source of the bearing problems.

Previously, when vibration analysis indicated that bearing condition was deteriorating, the motor would be replaced and sent to the shop for repair. We could predict, almost to the day, when the bearings on the 100 hp motors on certain presses would need to be changed. It was almost like clockwork. For one motor, it was about every 18 months, another was every 6 months, and still another every 2 years. Now we wanted to determine why they were failing.

When we disassembled a motor from the solvent recovery system, the bearing race looked like a washboard, which we knew indicated an electrical fluting problem. Disassembly of a motor in the press room revealed similar damage to the bearing.

Source of Electrical Fluting in Solvent Recovery System

The next problem was to find out what was causing the electrical fluting. A common source of problems in electrical motors can be the presence of electrical currents that flow across the rubbing surfaces of bearings. Knowing this could occur, we tested a motor in the solvent recovery system to determine whether some kind of voltage was traveling through this bearing.

The motor tested was installed on a blower, which operated at 1800 or 1785 rpm. The blower systems in the solvent recovery system had been modified, and variable frequency drives installed so that the motors would run at the air flow demand.

To test this particular motor, a motor brush was glued to the end of a piece of wood about a 1/4-inch thick by 1 inch wide, and approximately 1 foot long. One of the leads of a voltmeter was connected to the motor frame, which was electrically grounded and the other lead was attached to the motor brush. The reading showed a voltage of .4 to .5 volts. We also used a tape recorder, capable of taking up to 20 volts directly into the recorder, and recorded voltage directly on a tape.

The tape recorded data was analyzed. We found a voltage of 60 Hz, and harmonics thereof, coming through the bearing (Figure 1).

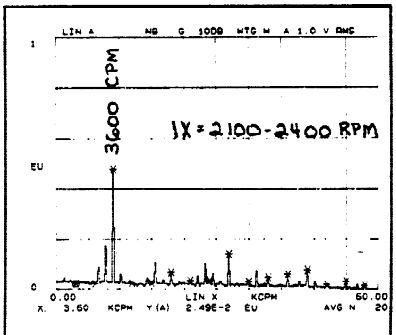

Figure 1

The motor was running at 35-40 Hz. The voltage is not the cause of the problem; it's the amperage going through the bearing. But we were on the right track.

At this point, electricians were called in to assist in identifying the source of the problem and to recommend solutions. With their assistance, we determined that in ing a variable frequency drive on a motor that was not designed to run with the drive created eddy currents within the motor. These currents were generated by the motor because of the frequency difference: instead of running at 60 Hz, we were running at 35-40 Hz, which generated electricity in the rotor that went through the bearings, through the frame, and back through the rear bearing. The electricity kept re-circulating in that fashion.

Understanding the Vibration Characteristics of Electrical Fluting on Recovery System

When eddy current electrical fluting occurs, the bearing gets fluted on the outer and inner races. Because it's an AC current, it actually takes the metal away from both races.

Generally, to identify this type of problem, most people first look for an outer or inner race defect. But the fluting or washboard effect on the bearing races is very minu the beginning you can hardly even feel ...

Even though there are defects in both races, the bearing problem cannot be detected by examining the bearing frequency, in this case, of 6500 cpm. At the most, we found

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that a tiny little spike might be observed, and this spike can occur even if you don't have a problem.

We learned that with electrical fluting, a spike that did not match any bearing frequency occurs between 40,000-60,000 cpm. Most often the spike was accompanied by side bands of whatever race had the biggest defect (Figure 2). It also could occur for both races. This was one way we could routinely identify the electrical fluting problems.

Another way to identify electrical fluting was listening to the motor itself. We noticed the motors sounded like they were under a heavy strain, yet the motors were under no load or under their normal load. As the bearing defect worsens, the sound intensifies and becomes deeper.

How Problem Was Corrected in Solvent Recovery System

The rear bearing in the motor was insulated to correct the problem. The rear bearing was selected because there is less load on that bearing than on the output shaft bearing driving the fan. The end bell of the motor was bored out, and a phenolic sleeve was installed. Then the bearing was pressed back into the sleeve, which prevents any electrical connection between the rear of the motor and the bell. Insulating the rear bearing eliminated the electrical fluting problems in the bearings.

Other Lessons Learned

During the time of insulating the rear bearings, we also installed power factor capacitors.

When power factor capacitors are the wrong size for the system, they actually generate

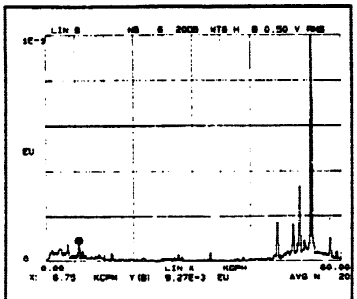

Figure 2

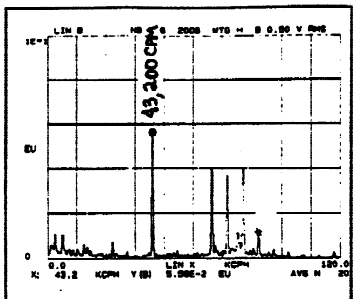

Figure 3

harmonics of line frequency, which are transmitted through the wires. It's like an intercom system you plug in the wall in one room to transmit music through the electrical outlet of another room. These harmonics were being sent across the line. We learned the power-factor capacitors have to be absolutely matched for the system.

Electrical Fluting on Motors in Press Room

Two 100 hp press drives with motors are installed on each press. The press drives utilize rectified AC current. We expected to see fluting on both bearing races when we disassembled the motor, just as we did with the solvent recovery system. But that was not the case. The fluting occurred only on one race, depending on the flow of current. The vibration spikes at 40,000-60,000 cpm were the same as on the motors in the solvent recovery system, with side bands of either the outer or inner race (Figure 3).

We have not identified the root source of the fluting problem on the press motors. We suspect that the harmonics of line frequency are being passed down through the motor, which generates a current in the motor shaft. These are not eddy currents, as they were in the solvent recovery system. The current in the press motors is generated in the rotor and goes directly through the bearings to ground. It doesn't circulate around.

We learned that the 5th, 7th and 9th harmonic of line frequency causes the problem. By recording the voltage on a tape recorder, we observed a badly distorted sine wave when

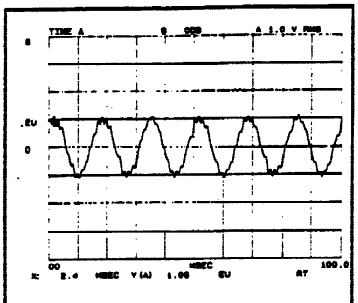

Figure 4

these harmonics are present. The sine wave is typically square and choppy on the top and bottom, rather than rounded (Figure 4). The worse the distortion, the worse the sine wave looks.

How Electrical Voltage Problem Was Corrected

To correct the problem, carbon brushes were installed on the the end of the rear bearings where the shaft goes through the bearing.

Two new chillers, with variable frequency drives, have been installed in the equipment mezzanine, and we have used the lessons learned from our previous experience to prevent similar electrical problems from occurring. The variable frequency drives and motors were designed to match to avoid creating the problem in the first place.

As an added precaution, filters were installed to prevent harmonics from reentering the system. As a rule, filters are installed on all new equipment.

A Point of Caution: A Remedy Is Not a Cure

Installing brushes on the back side of the motors in the press room have resolved the problem, but that does not mean the problem will not recur. Electricity takes the path of least resistance. If the brushes wear, and we do not detect it soon enough, the current will go back down through the bearings again.

While Update has not personally confirmed the accuracy of the preceding case history, we hope it is of interest to our readers and thank R.R. Donnelley for sharing their extensive experience in this area. Other companies have indicated similar problems with variable frequency drives. What have your experiences been?

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SIGNIFICANT HYDRAULIC FORCE VIBRATION IN A BOILER FEED PUMP

I. BACKGROUND

A fossil fuel utility company contracted Technical Associates to provide complete startup and implementation of a computer-based vibration program at each of their plants. Additionally, Technical Associates was to collect the initial baseline measurements and issue a baseline report on machine condition.

One of the machines included in the baselining was Boiler Feed Pump 681. This boiler feed pump was a turbine-powered variable speed machine outfitted with sleeve bearings on both the turbine and pump as seen in Figure 1. The latest vibration measurements from the baseline are shown in the Machine Report, Figure 2. Note that several pump positions were in alarm of overall vibration. Thus, the objective was to diagnose the cause of high vibration and recommend corrective actions if required.

II. RESULTS

Figures 3 thru 6 are spectra for the horizontal and vertical measurements at the inboard and outboard pump bearings (Positions 3 and 4 in the horizontal and vertical directions as shown in Figure 1). Note that each of the spectra contained harmonics of 5X RPM with noticeable amplitudes. The 5X RPM frequency was believed to be the blade passing frequency (BPF) of the pump even though the exact number of impeller blades was not known at that time. Figures 3, 4 and 6 were also dominated by a frequency of 100,350 CPM which was 15X RPM (or 3X BPF). Comparison of Figures 3 and 4 show the 100,350 CPM frequency had relatively the same amplitude in the horizontal and vertical directions with the vertical being slightly higher. However, comparison of Figures 5 and 6 of the outboard pump bearing showed the vertical amplitude at 100,350 CPM was many times higher than the horizontal which suggested that a vertical natural frequency close to 100,350 CPM might exist at the outboard pump bearing.

III. CONCLUSIONS

A. A flow related problem was indicated by the multiple harmonics of the probable blade passing frequency if the impeller(s) had 5 vanes. The flow problems may have been due to flow restrictions (elbows, valves, reducers, obstructions, etc.), eccentric positioning of the impeller in its housing and uneven gap between the rotating vanes and stationary diffuser vanes. The number of impeller vanes needed verified.

B. A vertical natural frequency at the probable 3X BPF of 100,350 CPM likely existed at the outboard (Position 4) pump bearing. A system natural frequency will sometimes coincide with the blade passing frequency (BPF) or its harmonics causing high vibration.

III. RECOMMENDATIONS

A. Verify the number of impeller vanes.

B. Continue to trend the BPF harmonics and the suspected natural frequency at all speeds as recommended in the baseline diagnostic report of Figure 7.

C. Track the amplitude of the frequency at 15X RPM as the turbine speed changes. If in fact there is a vertical natural frequency at or near 100,350 CPM there should be a significant amplitude change at 15X RPM, particularly at Position 4V (see Figures 1 and 6).

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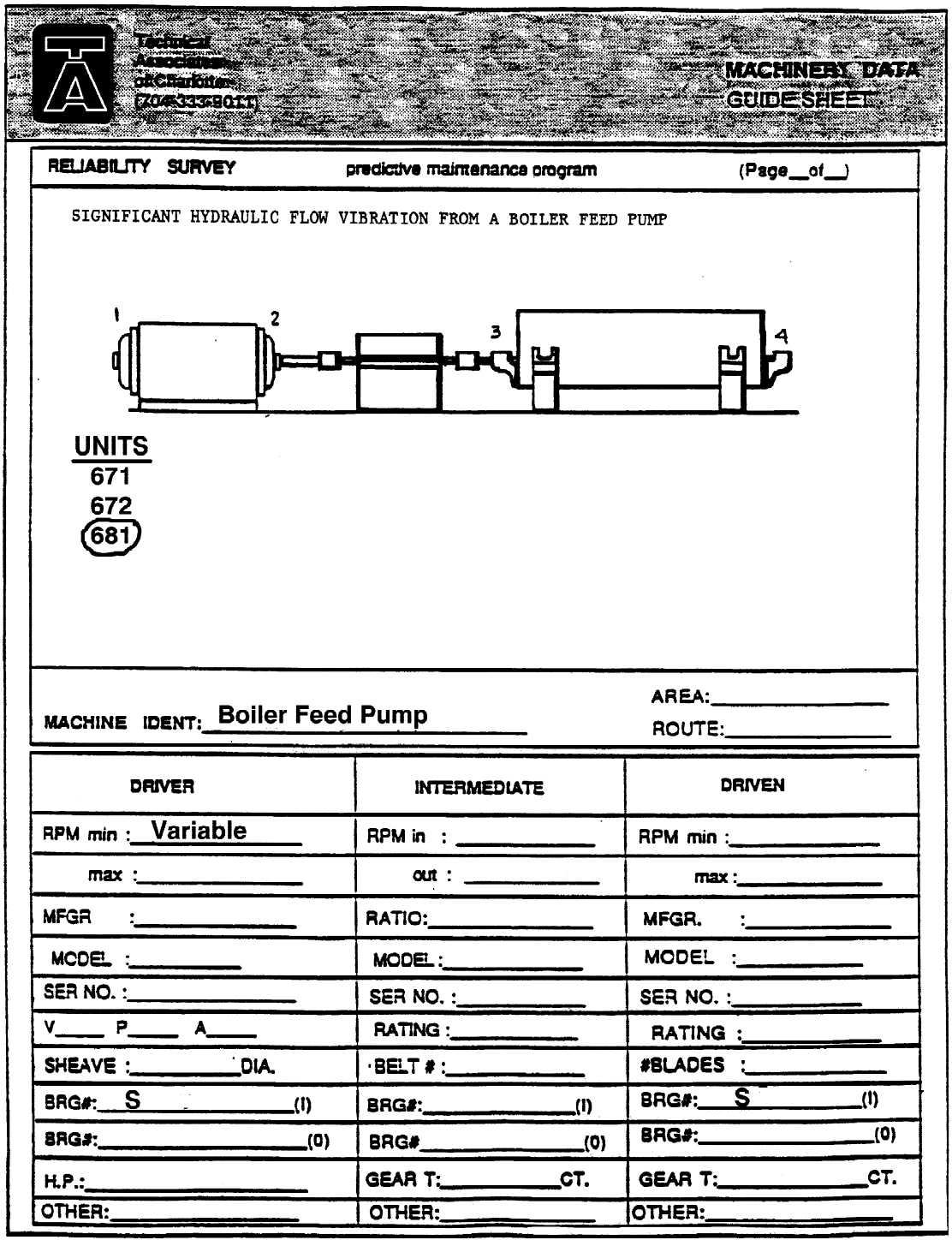

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MACHINERY DATA GUIDESHEET

RELIABILITY SURVEY — predictive maintenance program — (Page__of__)

SIGNIFICANT HYDRAULIC FLOW VIBRATION FROM A BOILER FEED PUMP

UNITS
671
672
(681)

MACHINE IDENT: Boiler Feed Pump

AREA:_____
ROUTE:_____

DRIVER	INTERMEDIATE	DRIVEN
RPM min : Variable	RPM in : _____	RPM min :_____
max :_____	out : _____	max :_____
MFGR :_____	RATIO:_____	MFGR. :_____
MODEL :_____	MODEL :_____	MODEL :_____
SER NO. :_____	SER NO. :_____	SER NO. :_____
V____ P____ A____	RATING :_____	RATING :_____
SHEAVE :_____DIA.	BELT # :_____	#BLADES :_____
BRG#: S_____(I)	BRG#:_____(I)	BRG#: S_____(I)
BRG#:_____(O)	BRG#_____(O)	BRG#:_____(O)
H.P.:_____	GEAR T:_____CT.	GEAR T:_____CT.
OTHER:_____	OTHER:_____	OTHER:_____

BFP - 04H

FIGURE 1

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IRD Mechanalysis Data Analysis System Page
Dec-04-89

Machine Report (Latest Data) for: BLR FD PUMP 681

Date	Time	Pos	Dir	Ampl	Units	Change	Alarm	Inspection
Aug-16-89	20:09	1	H	.099	IN/S	-----%	.300	
Aug-16-89	20:04	2	H	.107	IN/S	-----%	.300	
Aug-16-89	20:05	2	V	.181	IN/S	-----%	.300	
Aug-16-89	20:05	3	H	*.372	IN/S	-----%	.300	
Aug-16-89	20:05	3	V	*.451	IN/S	-----%	.300	
Aug-16-89	20:06	4	A	.177	IN/S	-----%	.300	
Aug-16-89	20:07	4	H	.176	IN/S	-----%	.300	
Aug-16-89	20:07	4	V	*.513	IN/S	-----%	.300	
Aug-16-89	20:28	RPM	0	6705	RPM	-----%	7000	

9 lines this report

FIGURE 2

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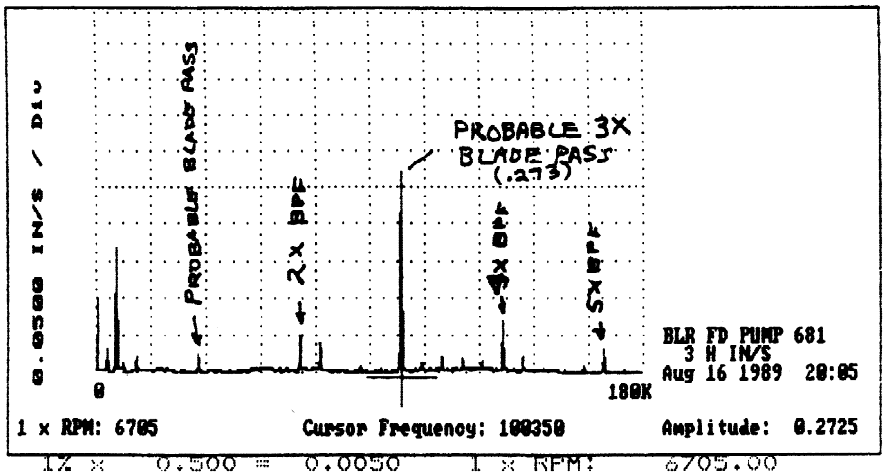

1% x 0.500 = 0.0050 1 x RPM: 6705.00

Machine	Pos	Dir	Units	Date
BLR FD PUMP 681	3	H	IN/S	Aug 16 1989 20:05

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
450	0.067	0.1000	133650	19.933	0.0700 -20x
3600	0.537	0.0300	140400	20.940	0.0225
6750	1.007	0.1700 - 1X	147150	21.946	0.0075
9000	1.342	0.0125	160200	23.893	0.0100
13500	2.013	0.0225 -2x	166950	24.899	0.0325 -25X
20250	3.020	0.0075 -3x			
33300	4.966	0.0250 - 5X			
55350	8.255	0.0075			
66600	9.933	0.0500 - 10X			
73350	10.940	0.0400			
86850	12.953	0.0100			
100350	14.966	0.2725 -15X			
107100	15.973	0.0125			
113400	16.913	0.0225			
120150	17.919	0.0200			
126900	18.926	0.0150			
128700	19.195	0.0075			

T - Threshold adjust Q - Quit

PLOT 22

FIGURE 3

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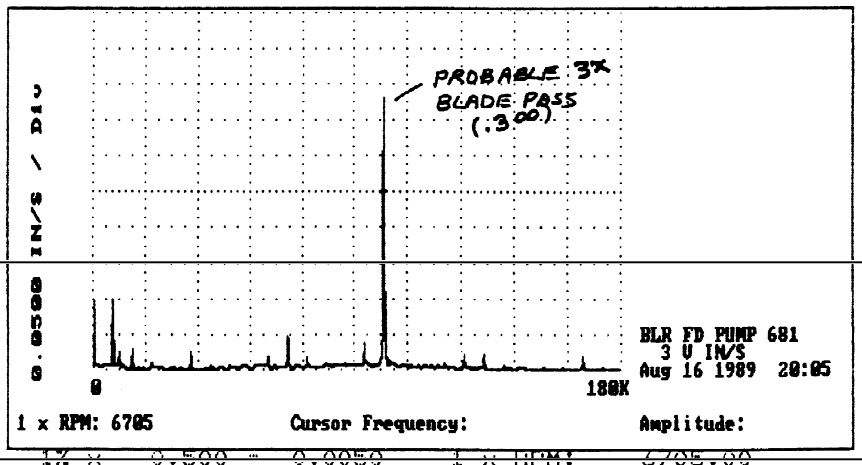

Machine	Pos	Dir	Units	Date
BLR FD PUMP 681	3	V	IN/S	Aug 16 1989 20:05

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitud
450	0.067	0.0975	86850	12.953	0.0100
~~3600~~	~~0.537~~	~~0.0075~~	~~88650~~	~~13.221~~	~~0.0075~~
6750	1.007	0.1000	89550	13.356	0.0075
9000	1.342	0.0250	91350	13.624	0.0075
13500	2.013	0.0275	93600	13.960	0.0375
20250	3.020	0.0100	100350	14.966	0.3 - 15X
33300	4.966	0.0250 - 5X			
46800	6.980	0.0075			
52650	7.852	0.0075			
56250	8.389	0.0075			
57600	8.591	0.0075			
60300	8.993	0.0200			
66600	9.933	0.0475 - 10X			
69750	10.403	0.0075			
73350	10.940	0.0175			
80100	11.946	0.0100			
84150	12.550	0.0075			

PLOT 23

FIGURE 4

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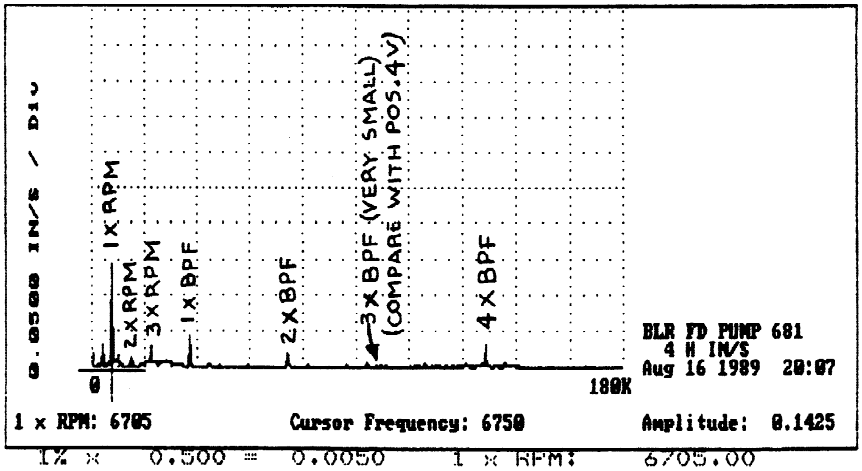

1% x 0.500 = 0.0050 1 x RPM: 6705.00

Machine	Pos	Dir	Units	Date
BLR FD PUMP 681	4	H	IN/S	Aug 16 1989 20:07

Frequency	1 x RPM	Amplitude	
450	0.067	0.0175	
3600	0.537	0.0300	
6750	1.007	0.1425	
9000	1.342	0.0150	
13500	2.013	0.0125	
20250	3.020	0.0275	
22050	3.289	0.0075	
25200	3.758	0.0100	
33300	4.966	0.0425	= 1X BPF = 5 X RPM
66600	9.933	0.0200	= 2X BPF = 10 X RPM
93600	13.960	0.0050	
113400	16.913	0.0050	
126900	18.926	0.0075	
132300	19.732	0.0050	
133650	19.933	0.0300	= 4X BPF = 20X RPM
140400	20.940	0.0050	

T - Threshold adjust Q - Quit

PLOT 24

FIGURE 5

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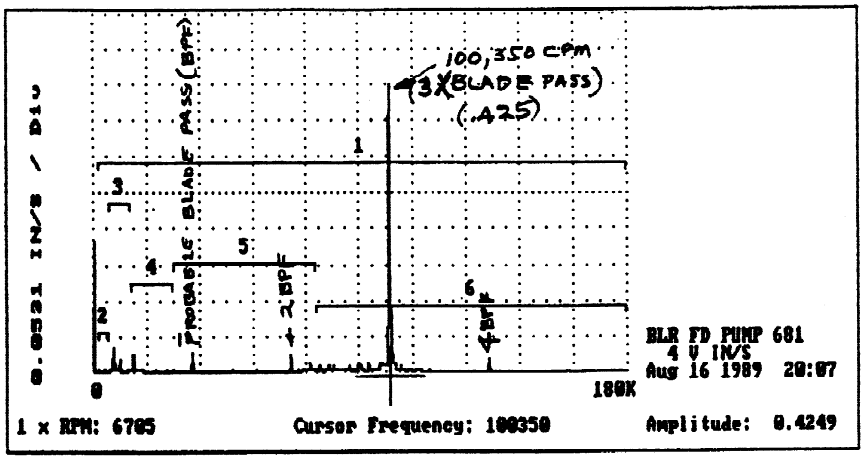

002% x 1.000 = 0.0200 1 x RPM: 6705.00

Machine	Pos	Dir	Units	Date
BLR FD PUMP 681	4	V	IN/S	Aug 16 1989 20:0

Frequency	1 x RPM	Amplitude
450	0.067	0.1951
6750	1.007	0.0349 - 1x
9000	1.342	0.0200
13500	2.013	0.0249 - 2x
33300	4.966	0.0300 - 5x
66600	9.933	0.0249 - 10x
100350	14.966	0.4249 - 15x
133650	19.933	0.0200 - 20x

T - Threshold adjust Q - Quit

PLOT 25

FIGURE 6

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SUMMARY OF RESULTS AND RECOMMENDATIONS FOR MACHINES FOUND TO HAVE PROBLEMS

(SORTED FIRST BY PROBLEM PRIORITY, THEN BY PROBLEM SEVERITY WITHIN PRIORITY)

AS OF: 8/18/80

AREA	MACHINE COMPUTER NAME	INDICATED PROBLEM(S): UNBALANCE	MISALIGNMENT/SOFT FOOT	BENT SHAFT	LUBRICATION PROBLEM	BEARING PROBLEM	MECHAN. LOOSENESS	BELT/SHEAVE PROBLEM	RESONANCE PROBLEM	GEAR PROBLEM	BLADE PASSING PROB.	CAVITATION	MOTOR ELECTRICAL	OIL WHIRL /WHIP	LOOSE FOUNDATION	VIB. MONITOR PROB.	SEAL /ROTOR RUB	OTHER (SEE REMARKS)	*PRIORITY OF PROBLEM	HIGHEST OVERALL VIBRATION: POSITION NO.	LEVEL (m = mils, i = in/sec, g = g's)	RECOMMENDATIONS & REMARKS	SEE PLOT NO.
	SLAG SLC PMP621					●													2	1H 2V	.211 i .259 i	Probable bearing defect frequencies (5550 CPM) seen at Positions 1H and 2V combined with harmonics of running speed indicating mechanical looseness. Spike energy (g-SE) also in alarm at 1H indicating bearing related problem. The mechanical looseness may be the result of bearing wear. Suggest replacing motor bearings. Positions 3H and 4H also exhibit possible bearing defect frequencies. Trend these in future surveys.	66 67 68
	ICVS FAN NO. 64																	●	3	1H	.544 i	Band 1 alarm due to a 600 CPM vibration from an unidentified source. An attempt should be made to identify the source. Trend in future surveys.	
	CONTROL RECP675	●	●					●											3	1A 1V	.454 i .526 i	Motor sheave misalignment, wobble and/or eccentricity with possible unbalance, as seen in Plots 31 and 32. Trend in future surveys.	31 32
	PULVERIZER 662					●			●									●	3	1A 1V	.464 i .520 i	Band alarms at Position 1A and 1V due to a broad band vibration from 3060 CPM to 3660 CPM possibly due to a bearing problem, resonance or vibration transmittal from another source, possibly the pulverizer itself. Suggest further diagnostics to determine the dominant pulverizer frequencies.	61 62
	BLR FD PUMP 681								●		●							●	3	3H 3V 4H 4V	.285 i .451 i .176 i .513 i	Flow related problem indicated by high three times passing frequency at 10,350 CPM. If the impeller has five vanes, as indicated by Plots 22, 23 & 25. However, comparing horizontal and vertical Plots 24 and 25 from Position 4 leads one to believe that resonance may exist. Verify number of impeller blades. Trend in future surveys.	22 23 24 25

① = TAKE **IMMEDIATE** REPAIR ACTION DUE TO SEVERE PROBLEM(S) NOTED (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
2 = SCHEDULE REPAIRS TO CORRECT PROBLEMS NOTED AT CONVENIENCE (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
3 = DESIGNATED PROBLEM DOES NOT WARRANT ACTION AT THIS TIME. (CONTINUE ONLY WITH RELIABILITY SURVEY TREND ANALYSIS)

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(704) 333-9011

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FIGURE 7

CAVITATION DIAGNOSIS WITHIN A CHILLED WATER PUMP

I. BACKGROUND

One of Technical Associates' predictive maintenance annual contract clients has had cavitation problems on a number of their pumps for many years. Apparently, the pumping systems were either incorrectly designed initially, or changes had been made which detrimentally affected the pumping system and resulted in cavitation. The purpose of this investigation was to confirm the real problem source and make remedial recommendations.

II. RESULTS

The difficulty with cavitating pumps is to confirm if the condition is actually cavitation which damages pump internals over time, or if it is a bearing problem with more immediate consequences. CARR 4 CHILLWPMP (Carrier 4 Chill Water Pump) was such a pump. The pump bearings' spike energy overall level and velocity band alarms would go in and out of alarm for what appeared to be an anti-friction bearing problem, but in reality, was cavitation.

Figure 1 illustrates the CARR 4 CHILLWPMP arrangement along with a typical spectrum taken on the inboard pump bearing having a raised baseline, along with a few spikes that could be either a bearing defect frequency (or harmonic) or blade passing frequency harmonics. Note this broadband energy predominantly between 14,000 and 44,000 CPM in the Figure 1 spectrum which can either indicate cavitation or the final stage of the life of a rolling element bearing. Figure 2 shows the spike energy trend on the lower graph with velocity spectra in the upper two plots for all directions (A, H and V) at the two pump bearings (Positions 3 and 4). Neither seemed to help in solving the dilemma of determining if the problem was cavitation or bearing wear.

What did help was to look at the waterfall velocity spectra of Figures 3 and 4. All the waterfall spectra at each point have had the same general spectral appearance for at least a year and more. If the problem were due to bearing deterioration, it would have failed or at least worsened with this machine operating 365 days per year. Since no real change was evident between the spectra, cavitation was the likely problem.

Another help is that generally two bearings on the same shaft do not fail at the same rate at the same time. As in this case, both bearings could not remain at the same deterioration state for over a year. Another strong clue indicating cavitation was the sound of the pump. Cavitation will have a "sandy" to "gravely" sound as if the machine were pumping a dense sand or gravel slurry instead of a clean liquid. In this case, such a noise was documented by the analyst several times on his monthly routes.

One other possible problem that could be present in this situation is a lack of lubrication. Spike energy measurements are sensitive to unlubricated anti-friction bearing component movements. However, were it a lubrication problem, the pump bearings could not have continued to operate for nearly two years without fatigue failure.

III. CONCLUSIONS

A. The problems detected by the velocity band alarms and spike energy alarms were diagnosed to be caused by cavitation.

B. Left uncorrected, such cavitation will eventually cause great damage to the internals of the pump, and even to the connected piping.

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IV. RECOMMENDATIONS

A. Take action to confirm the source of the continuing cavitation problem and take corrective actions to eliminate it. Ensure the valves are working properly; that no obstructions are present within the suction piping; and also ensure it is not being operated beyond its performance capabilities. Prolonged cavitation will result in internal pump damage.

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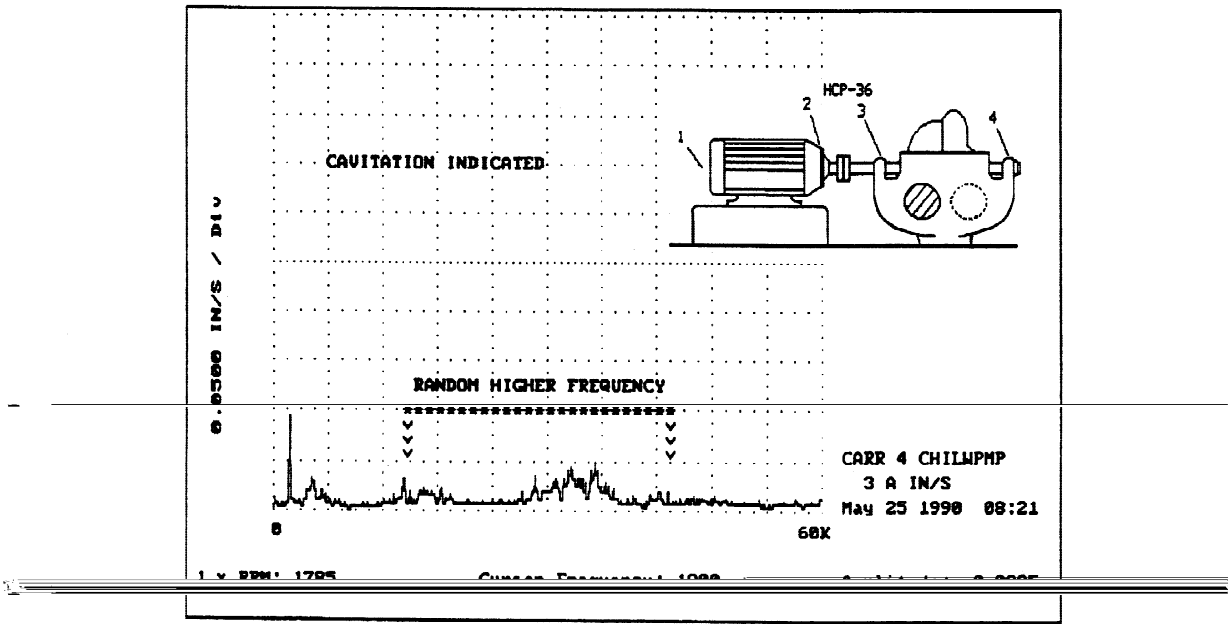

2% x 0.500 = 0.0100 1 x RPM: 1785.00

Machine	Pos	Dir	Units	Date
CARR 4 CHILWPMP	3	A	IN/S	May 25 1990 08:21

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
1800	1.01	0.0925	32550	18.24	0.0425
4200	2.35	0.0300	33450	18.74	0.0325
5400	3.03	0.0200	33900	18.99	0.0300
14400	8.07	0.0325	34200	19.16	0.0275
15000	8.40	0.0175	34800	19.50	0.0400
16200	9.08	0.0200	35250	19.75	0.0500
16500	9.24	0.0225	35550	19.92	0.0400
16950	9.50	0.0175	36450	20.42	0.0275
17400	9.75	0.0175	36750	20.59	0.0250
18450	10.34	0.0225	37350	20.92	0.0225
27300	15.29	0.0175	42300	23.70	0.0175
28650	16.05	0.0350	43350	24.29	0.0175
29850	16.72	0.0200			
30600	17.14	0.0275			
30900	17.31	0.0325			
31800	17.82	0.0325			
32250	18.07	0.0400			

T - Threshold adjust Q - Quit

FIGURE 1

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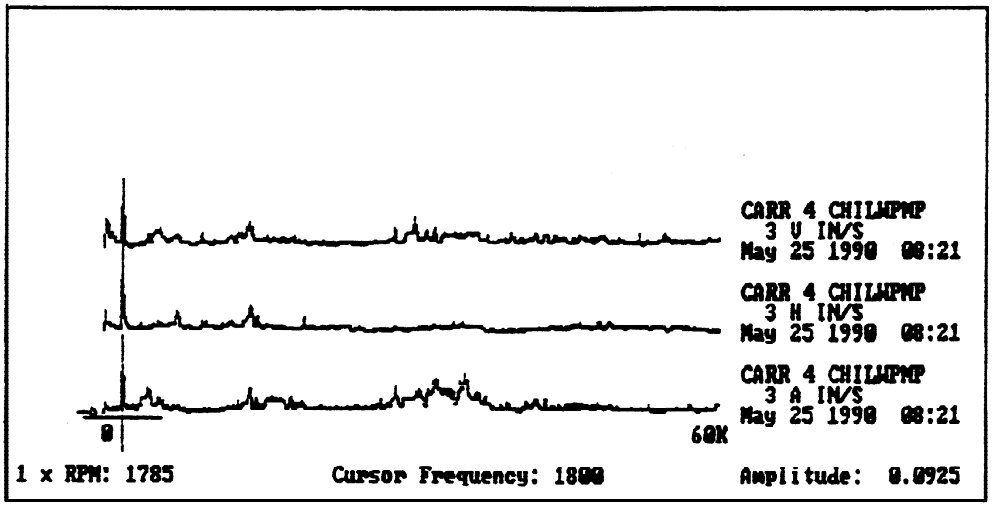

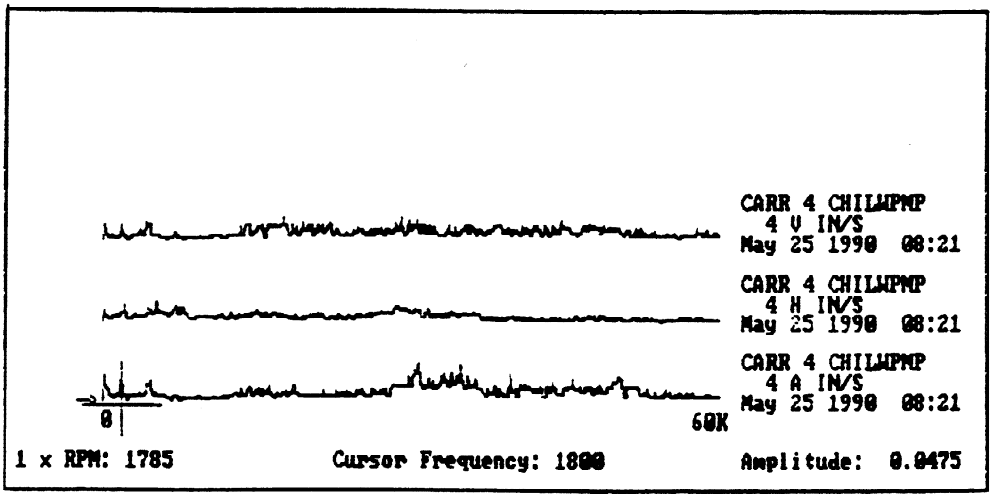

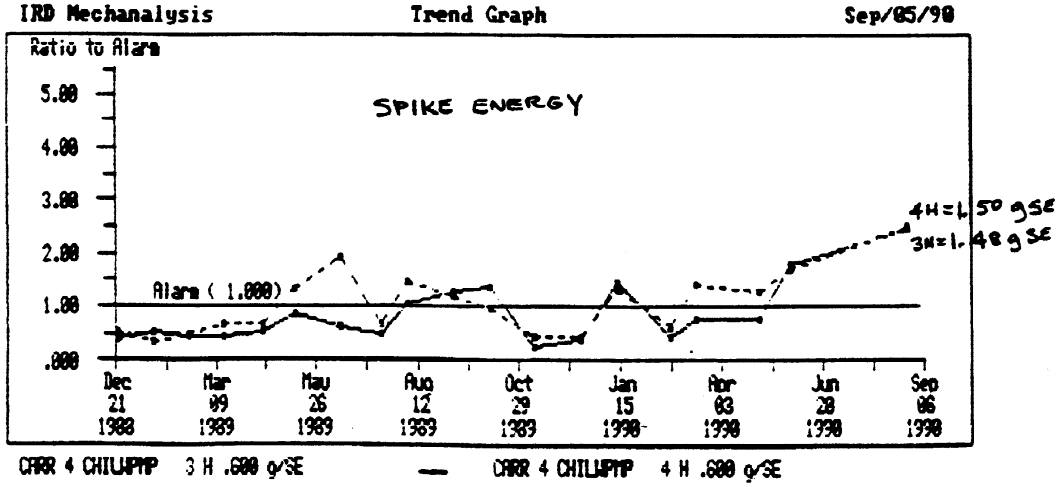

FIGURE 2

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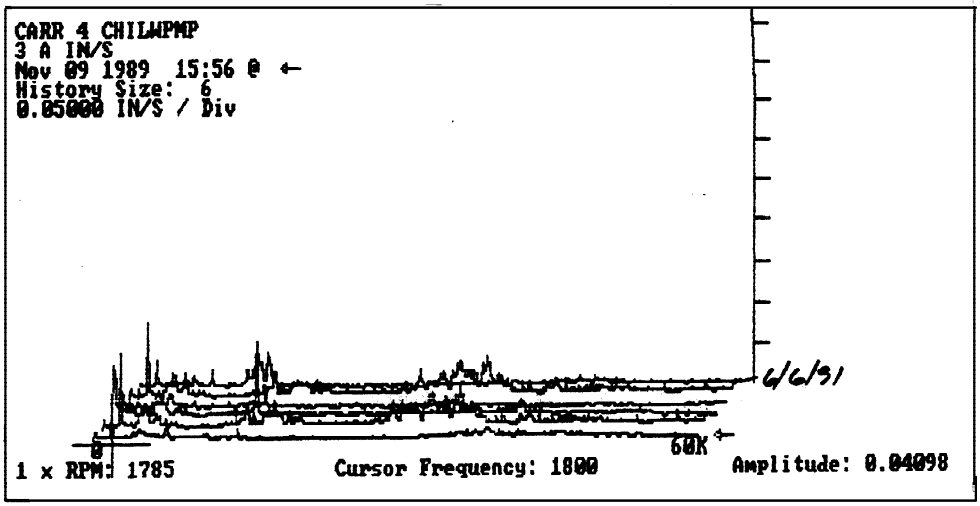

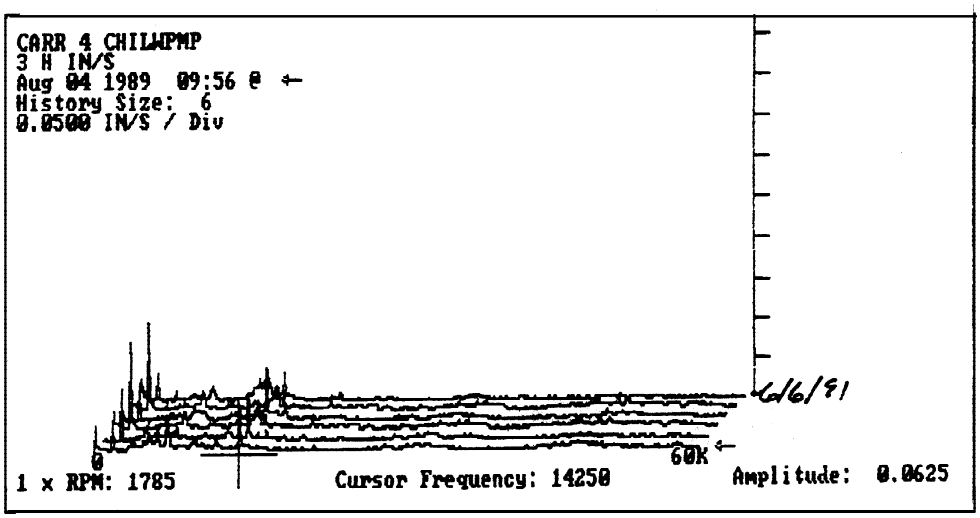

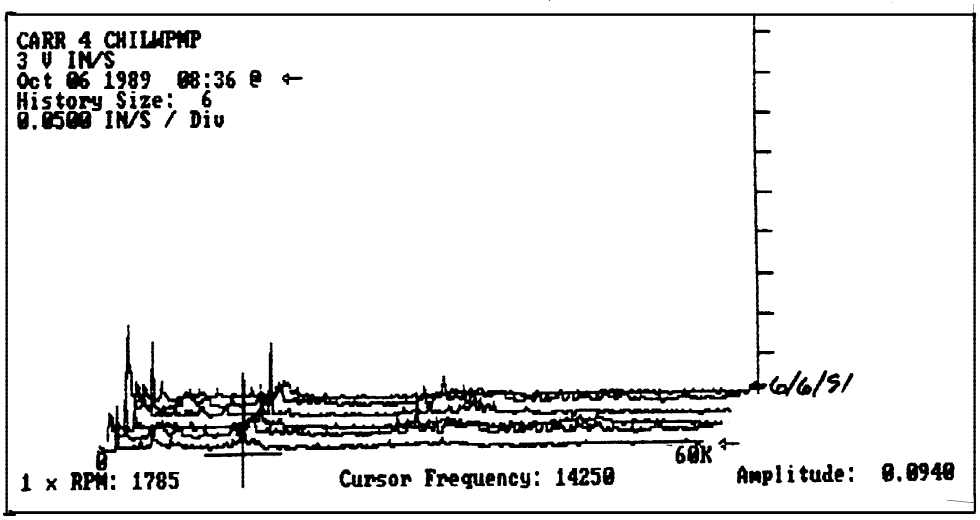

FIGURE 3

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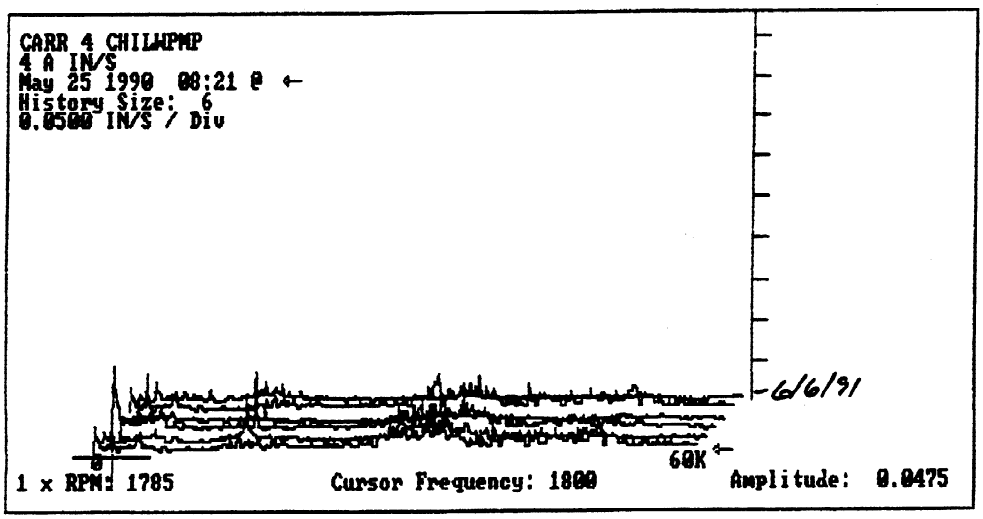

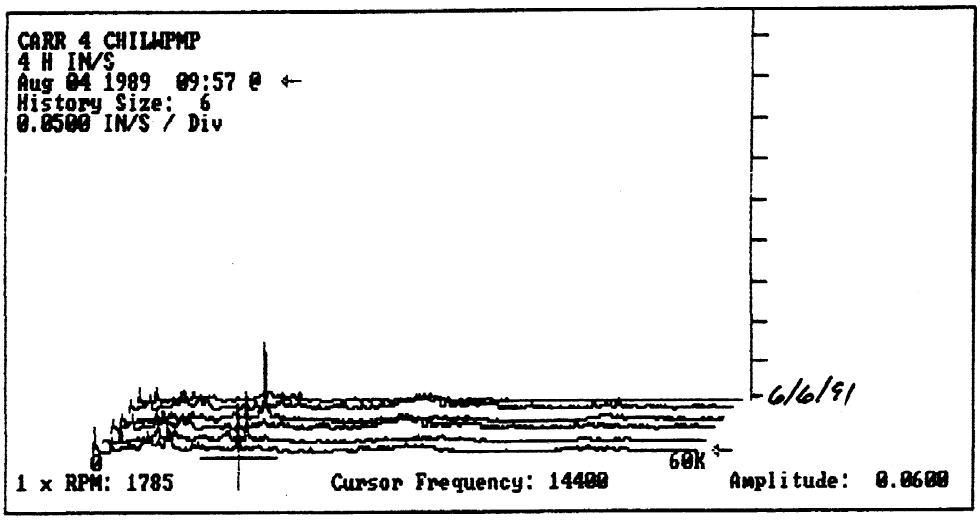

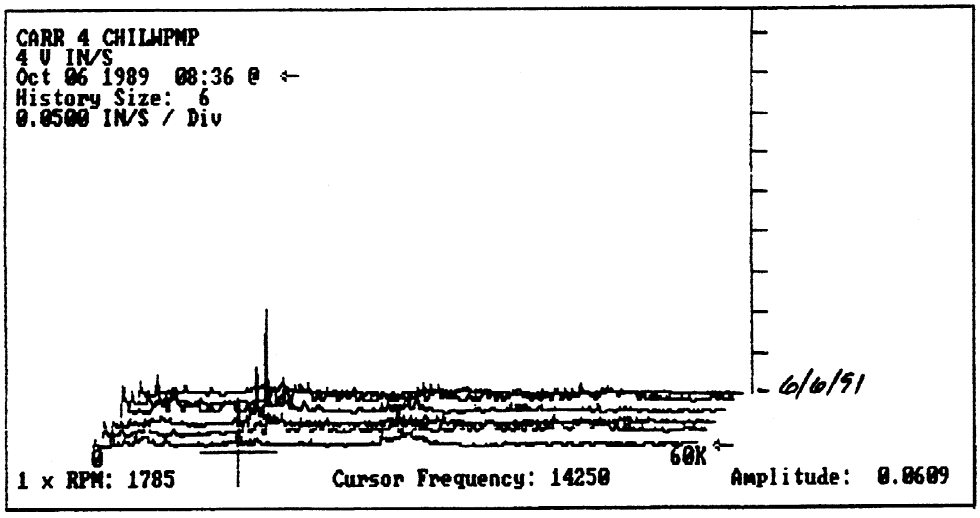

FIGURE 4

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DISTINGUISHING BETWEEN PUMP CAVITATION & DEFECTIVE BEARINGS

I. BACKGROUND

This case history is about a pump monitored by Technical Associates on one of our annual contract Predictive Maintenance Programs. It addresses one of the most important diagnostic tasks which must be performed on centrifugal pumps outfitted with rolling element bearings - distinguishing between cavitation and defective bearings (recall that cavitation normally causes random broadband vibration in the 20K - 120K CPM range as can the final failure stage of rolling element bearings). A machine drawing with position locations is shown in Figure 1. In this example, 344 HORPUMP 002 triggered a Band 6 alarm as seen in Figure 2 (the total RSS energy within Band 6 exceeded its alarm although no single frequency surpassed this level). The alarm was at Position 3V on the inboard pump bearing. The random, broadband high frequency vibration shown in Figure 2 gives the appearance of both cavitation and the final stages of bearing wear. The spikes superimposed in the broadband region in the January 31, 1991 spectra could be caused by either blade passing frequency harmonics or bearing natural frequency with 1X RPM sidebands. The number of impeller blades and bearing model number were not known. Spike energy levels at this time were .164 gSE at Position 3H and .125 gSE at Position 4H. At this time, the maintenance people were told the condition could be cavitation or a possible bearing problem.

Two months later on March 27, 1991, the problem developed into a well defined bearing defect. Spike energy was now .720 gSE at Position 3H and .516 at Position 4H. The spectra for both Position 3H and Position 4H are shown in Figures 3 and 4. Figure 5 is a waterfall of spectra captured on Position 3H. The possibility of cavitation was still there, but the dominant problem was now defective bearings. Both bearings now had multiple harmonics of a 6300 CPM frequency. The 6300 CPM frequency is a non-integer multiple of RPM and typical of a bearing defect (i.e., outer race, inner race or ball). A request for maintenance personnel to change the bearings at this time was ignored.

By May 15, 1991, after another two (2) months had passed, the spike energy was now 1.10 gSE at Position 3H and 0.97 gSE at Position 4H, and the spectrum for Position 3H deteriorated to that shown in Figure 6 (note that the peak at 4X Defect Frequency increased from .142 in/sec on March 27 up to .300 in/sec on May 15 with many other peaks growing as well). The bearings had now lasted four (4) months without failing catastrophically, but their life was extremely precarious. A strong plea was made to the maintenance department who finally relented and replaced the bearings. The result is shown in Figures 7 and 8 for Positions 3H and 4H. Spike energy values were reduced to .175 gSE and .251 gSE for Positions 3H and 4H (still not down to the levels below .05 to .06 gSE for which we had hoped). The new spectra showed continued evidence of cavitation with the low broadband energy area, and still exhibited what might be interpreted as bearing wear. However, the most likely problem was cavitation due to the similar gSE levels as well as the similar spectra at Positions 3H and 4H (Figures 7 & 8). In addition, the plant made no attempts to check net positive suction head (NPSH), nor made any changes which would affect fluid flow at the pump inlet. The key point is, late stages of bearing defects and cavitation are similar in appearance and must be monitored carefully to see if the condition will develop into a bearing problem.

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II. CONCLUSIONS

1. Cavitation and anti-friction bearing problems can be similar in appearance.

2. Obtain as much information about the machine being monitored as possible (i.e., bearing model numbers, number of impeller vanes, etc.) to enhance diagnostic capabilities.

3. The increasing spike energy (gSE) values were indicative of the deterioration of the anti-friction bearings.

III. RECOMMENDATIONS

1. Once a cavitation-like condition is discovered in a pump, it needs to be monitored closely to detect any anti-friction bearing problems that may develop.

2. When the problem is proven to be bearing wear, replace the bearings as soon as possible, not allowing them to approach catastrophic failure as did the pump in this case history.

3. In addition to replacing the bearings, look for the root cause(s) that might have contributed to bearing problems. In this case, if cavitation is suspected, determine if flow can be altered in such a way to eliminate (or at least minimize) the cavitation problem.

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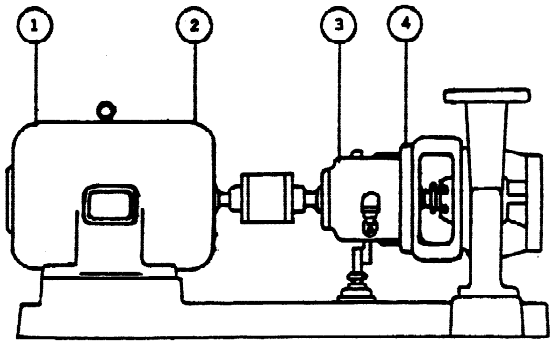
Technical Associates of Charlotte (704-333-9011)

MACHINERY DATA GUIDE SHEET

RELIABILITY SURVEY | predictive maintenance program | (Page__of__)



SIDE VIEW

MACHINE IDENT: 344 Hor Pump 002

AREA: _____

ROUTE: _____

DRIVER	INTERMEDIATE	DRIVEN
RPM min : 1750 RPM	RPM in : _____	RPM min : 1750 RPM
max : _____	out : _____	max : _____
MFGR : _____	RATIO: _____	MFGR. : _____
MODEL : _____	MODEL : _____	MODEL : _____
SER NO. : _____	SER NO. : _____	SER NO. : _____
V____ P____ A____	RATING : _____	RATING : _____
SHEAVE : _____ DIA.	BELT # : _____	#BLADES : Unknown
BRG#: Unknown (I)	BRG#: _____ (I)	BRG#: Unknown (I)
BRG#: Unknown (0)	BRG# _____ (0)	BRG#: Unknown (0)
H.P.: _____	GEAR T: _____ CT.	GEAR T: _____ CT.
OTHER: _____	OTHER: _____	OTHER: _____

HCP-2

FIGURE 1

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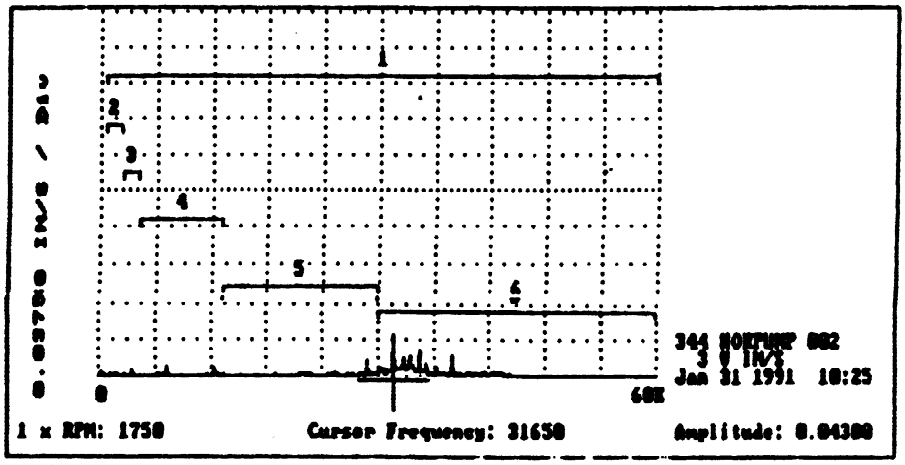

002X x 0.1000 = 0.00200 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
344 HORPUMP 002	3	V	IN/S	Jan 31 1991 10:

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
150	0.09	0.00550	34500	19.71	0.02550
1800	1.03	0.00300	35100	20.06	0.01299
3450	1.97	0.00701	35550	20.31	0.00500
6300	3.60	0.00249	36300	20.74	0.00999
7200	4.11	0.00950	36900	21.09	0.00349
12300	7.03	0.00950	37950	21.69	0.02200
25350	14.49	0.00550	38700	22.11	0.00249
28800	16.46	0.01550	39750	22.71	0.00500
29250	16.71	0.00400	40050	22.89	0.00300
29850	17.06	0.00500	41550	23.74	0.00450
30300	17.31	0.00850	42600	24.34	0.00400
30600	17.49	0.00701	43200	24.69	0.00450
30900	17.66	0.00650	44250	25.29	0.00300
31650	18.09	0.04300			
32700	18.69	0.01900			
33450	19.11	0.02200			
33750	19.29	0.01099			

T - Threshold adjust Q - Quit

FIGURE 2

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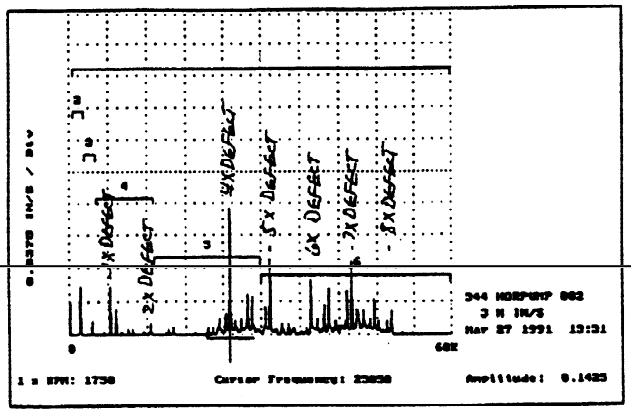
FIGURE 3



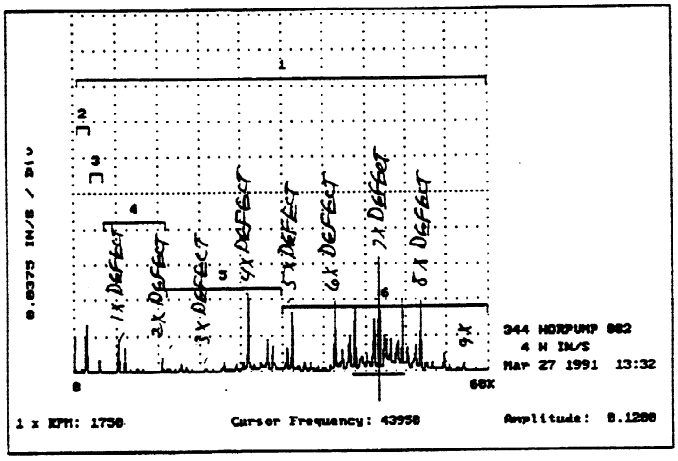
002% x 0.500 = 0.0100 1 x RPM: 1750.00

Machine: 344 HORPUMP 002 | Pos: 3 | Dir: H | Units: IN/S | Date: Mar 27 1991 13:31

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
150	0.09	0.0350	37650	21.51	0.0600
1800	1.03	0.0550	38700	22.11	0.0175
3600	2.06	0.0150	39750	22.71	0.0325
6300	3.60	0.0550	40500	23.14	0.0525
7200	4.11	0.0275	41550	23.74	0.0150
12600	7.20	0.0125	42450	24.26	0.0100
23400	13.37	0.0175	43200	24.69	0.0500
24450	13.97	0.0225	43950	25.11	0.0825
25050	14.31	0.1425	44700	25.54	0.0275
25950	14.83	0.0150	45750	26.14	0.0250
26850	15.34	0.0175	46800	26.74	0.0175
27900	15.94	0.0450	47400	27.09	0.0400
28650	16.37	0.0425	48150	27.51	0.0150
30600	17.49	0.0300	49500	28.29	0.0150
31350	17.91	0.0850	50250	28.71	0.0250
33150	18.94	0.0125			
34200	19.54	0.0125			

T - Threshold adjust Q - Quit

FIGURE 4



002% x 0.500 = 0.0100 1 x RPM: 1750.00

Machine: 344 HORPUMP 002 | Pos: 4 | Dir: H | Units: IN/S | Date: Mar 27 1991 13:32

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
150	0.09	0.0375	43200	24.69	0.0550
1800	1.03	0.0500	43950	25.11	0.1200
3600	2.06	0.0125	44700	25.54	0.0375
6300	3.60	0.0325	45000	25.71	0.0325
7200	4.11	0.0225	45750	26.14	0.0325
12600	7.20	0.0150	46800	26.74	0.0300
25050	14.31	0.0825	47400	27.09	0.0750
27900	15.94	0.0325	48300	27.60	0.0225
28650	16.37	0.0250	49500	28.29	0.0250
30600	17.49	0.0225	50250	28.71	0.0725
31350	17.91	0.0600	53700	30.69	0.0175
37650	21.51	0.0725			
38700	22.11	0.0200			
39750	22.71	0.0375			
40500	23.14	0.0675			
41550	23.74	0.0150			
42150	24.09	0.0175			

T - Threshold adjust Q - Quit

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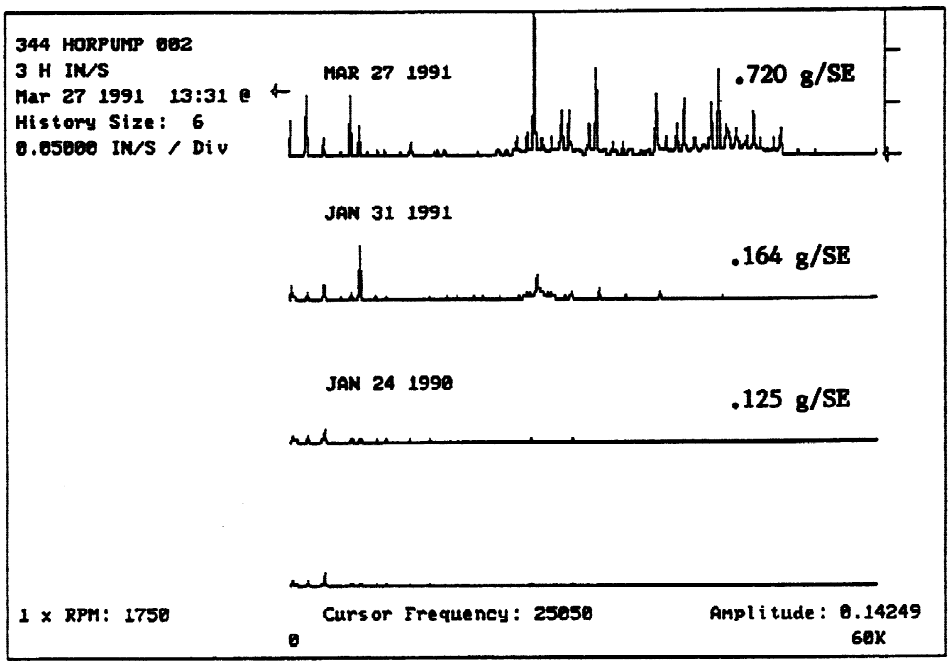

1% x 0.5000 = 0.00500 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
344 HORPUMP 002	3	H	IN/S	Mar 27 1991 13:31

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
150	0.09	0.03503	37650	21.51	0.05997
1800	1.03	0.05496	38700	22.11	0.01747
3600	2.06	0.01501	39750	22.71	0.03249
6300	3.60	0.05496	40500	23.14	0.05250
7200	4.11	0.02748	41550	23.74	0.01501
12600	7.20	0.01247	42450	24.26	0.01001
23400	13.37	0.01747	43200	24.69	0.04996
24450	13.97	0.02248	43950	25.11	0.08253
25050	14.31	0.14249	44700	25.54	0.02748
25950	14.83	0.01501	45750	26.14	0.02502
26850	15.34	0.01747	46800	26.74	0.01747
27900	15.94	0.04504	47400	27.09	0.04003
28650	16.37	0.04249	48150	27.51	0.01501
30600	17.49	0.03002	49500	28.29	0.01501
31350	17.91	0.08499	50250	28.71	0.02502
33150	18.94	0.01247			
34200	19.54	0.01247			

T - Threshold adjust Q - Quit

FIGURE 5

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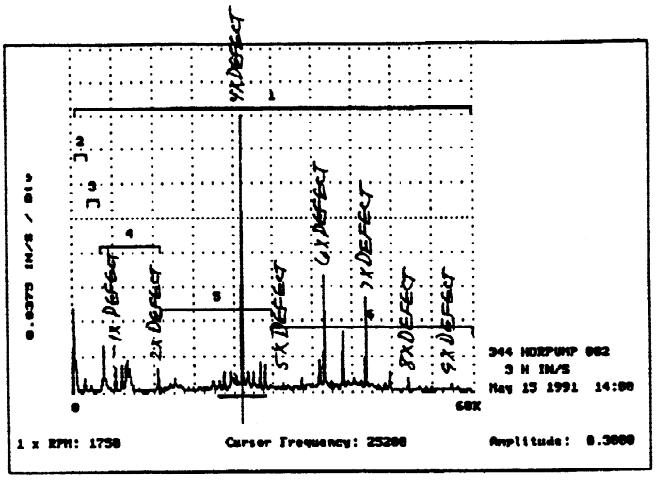


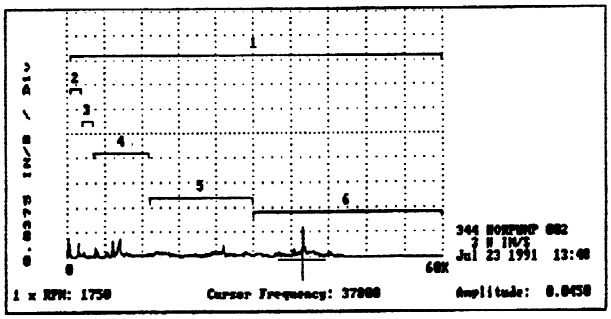
002% x 0.500 = 0.0100 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
344 HORPUMP 002	3	H	IN/S	May 15 1991 14:08

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
150	0.09	0.0900	27000	15.43	0.0200
1800	1.03	0.0150	28050	16.03	0.0300
4500	2.57	0.0500	28800	16.46	0.0275
6300	3.60	0.0250	34350	19.63	0.0150
7200	4.11	0.0275	37050	21.17	0.0325
8100	4.63	0.0325	37800	21.60	0.1250
8400	4.80	0.0225	38850	22.20	0.0125
12600	7.20	0.0225	40650	23.23	0.0625
15150	8.66	0.0150	41400	23.66	0.0125
20850	11.91	0.0125	42300	24.17	0.0150
21750	12.43	0.0125	43350	24.77	0.0125
22500	12.86	0.0200	44100	25.20	0.1000
23400	13.37	0.0200	47700	27.26	0.0200
24300	13.89	0.0175	50400	28.80	0.0150
24600	14.06	0.0125			
25200	14.40	0.3000			
26100	14.91	0.0200			

T - Threshold adjust Q - Quit

FIGURE 6



002% x 0.200 = 0.0040 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
344 HORPUMP 002	3	H	IN/S	Jul 23 1991 13:4

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
150	0.09	0.0280	30450	17.40	0.0050
1800	1.03	0.0220	30900	17.66	0.0050
2700	1.54	0.0060	34050	19.46	0.0060
4500	2.57	0.0140	34500	19.71	0.0050
6300	3.60	0.0070	36450	20.83	0.0120
7200	4.11	0.0240	37350	21.34	0.0060
8400	4.80	0.0250	37800	21.60	0.0450
9450	5.40	0.0060	38250	21.86	0.0140
14100	8.06	0.0080	39150	22.37	0.0080
14400	8.23	0.0080	39600	22.63	0.0090
15150	8.66	0.0070	42300	24.17	0.0060
15750	9.00	0.0060			
16350	9.34	0.0070			
23550	13.46	0.0070			
24450	13.97	0.0060			
25200	14.40	0.0170			
28650	16.37	0.0070			

T - Threshold adjust Q - Quit

FIGURE 7

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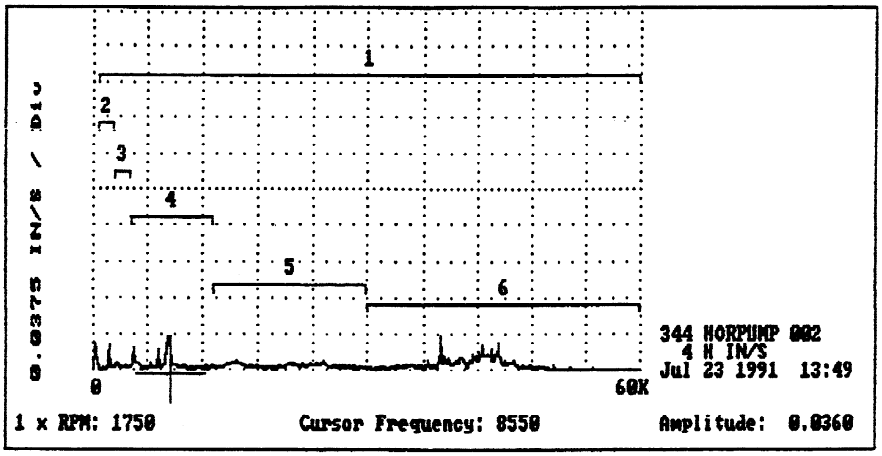

002% x 0.200 = 0.0040 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
344 HORPUMP 002	4	H	IN/S	Jul 23 1991 13:49

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
150	0.09	0.0280	39600	22.63	0.0100
1800	1.03	0.0260	40200	22.97	0.0120
2550	1.46	0.0080	41100	23.49	0.0090
4500	2.57	0.0240	41400	23.66	0.0160
6300	3.60	0.0070	41850	23.91	0.0150
7200	4.11	0.0220	42150	24.09	0.0170
8250	4.71	0.0350	42450	24.26	0.0250
8550	4.89	0.0360	43050	24.60	0.0170
15300	8.74	0.0080	43500	24.86	0.0210
15750	9.00	0.0090	43800	25.03	0.0160
21450	12.26	0.0070	44250	25.29	0.0270
24900	14.23	0.0080	45000	25.71	0.0090
25200	14.40	0.0100	45450	25.97	0.0070
36450	20.83	0.0090	45900	26.23	0.0100
37800	21.60	0.0340			
38250	21.86	0.0140			
38700	22.11	0.0110			

T - Threshold adjust Q - Quit

FIGURE 8

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DETECTION OF A GEAR WEAR PROBLEM ON A PAPER MACHINE

I. BACKGROUND

On May 2 and 3, 1990, a study of Paper Machine 18 Afterdrier was conducted to determine overall condition. Unbalance of some of the felt rolls was found but a potentially serious problem in the gearing was found that appeared to be a loose pinion. Maintenance work performed during July found a loose pinion and a loose pinion bearing. Follow-up measurements on July 25 were conducted by Technical Associates to determine whether the problem had been corrected. Figures 11.10A, 11.10B and 11.10C illustrate the machine configuration.

II. CONCLUSIONS/RECOMMENDATIONS

Vibration measurements of May 2 on the drive motor show that the dominant vibration on the motor was not at motor speed but at gear mesh frequency, as shown by Figure 11.10D. Figure 11.10A shows Position 1 on the motor where the Figure 11.10D vibration measurement was obtained. Figures 11.10E and 11.10F show how the vibration spectrum was saturated with harmonics of pinion RPM and that Gear Mesh Frequency had pinion RPM sidebands. The same measurement was obtained on July 25, after replacing the worn pinion. Figure 11.10G compares the vibration on the motor BEFORE and AFTER the pinion replacement. Note that the AFTER plot is only half of the full scale setting of the BEFORE plot, so the difference is even more dramatic than at first glance. Figure 11.10G shows clearly that the pinion problem that was corrected had a dramatic impact on machine reliability.

Table 11.10A below summarizes the axial vibration at gear mesh frequency for each pinion location, as numbered on Figure 11.10B, for the BEFORE and AFTER measurements:

TABLE 11.10 A

PINION VIBRATION AT GEAR MESH BEFORE & AFTER PINION REPLACEMENT (IN/SEC)

PINION NO.	VIBRATION AT GEAR MESH		% DIFFERENCE
	BEFORE	AFTER	
8	.196	.097	-50.5
9	.454	.258	-43.2
10	.434	.065	-85.0
11	.258	.103	-60.1
16	.310	.159	-48.7
17	.312	.135	-56.7
18	.525	.174	-66.8
19	.818	.191	-76.6
20	.344	.105	-69.5
21	.125	.232	+85.6

A significant reduction in gear mesh vibration at all but pinion 21 took place. The cause of the earlier reported pinion looseness has been solved.

III. RESULTS

The pinion looseness which threatened machine reliability has been corrected. A significant reduction in gear mesh vibration has taken place since the maintenance work of July 25, particularly in the vicinity of the main drive pinion, which was the location of the loose pinion. The dominant vibration at the drive motor was at gear mesh frequency prior to the maintenance work. Following the maintenance work, gear mesh vibration at the motor is almost completely gone.

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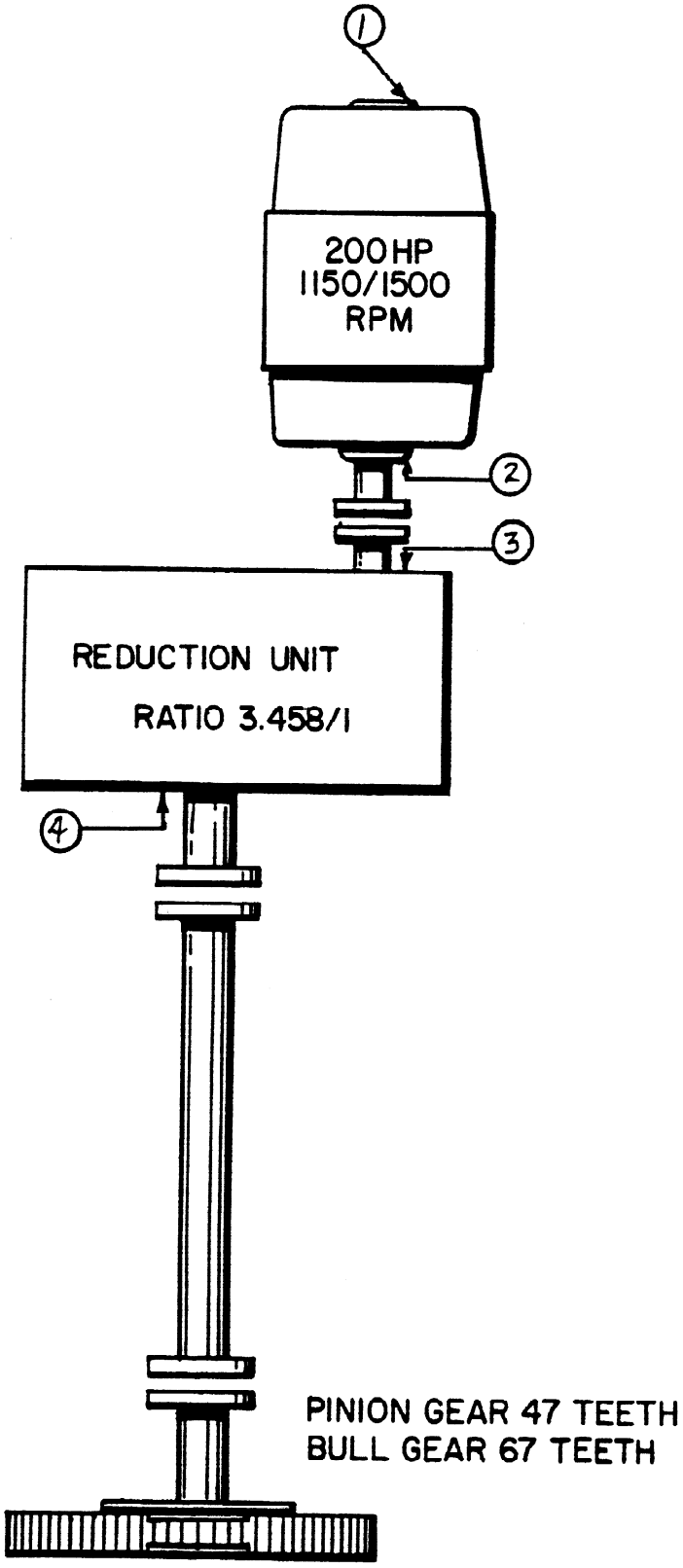

FIGURE 11.10-A

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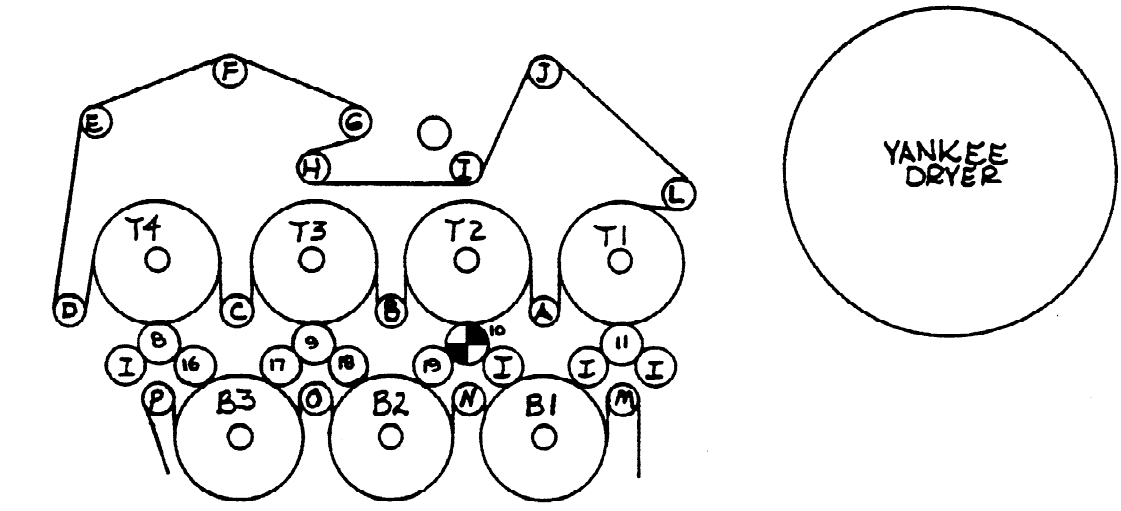

FIGURE 11.10-B

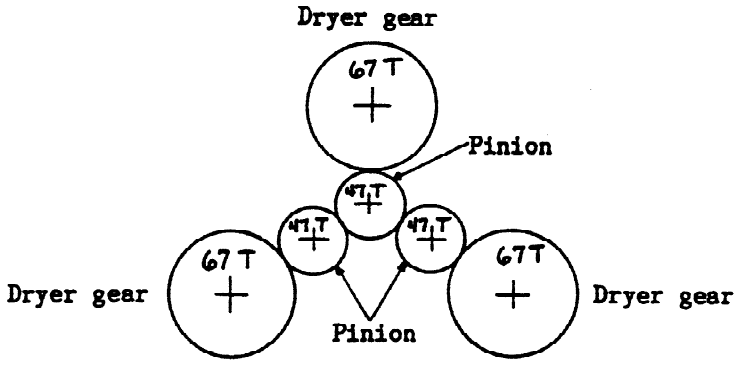

FIGURE 11.10-C

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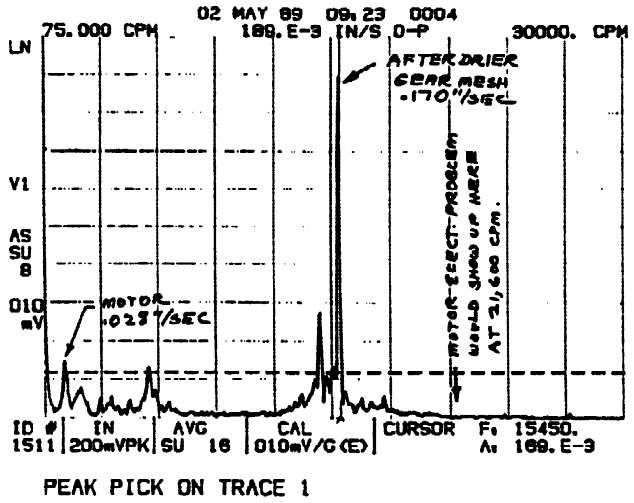
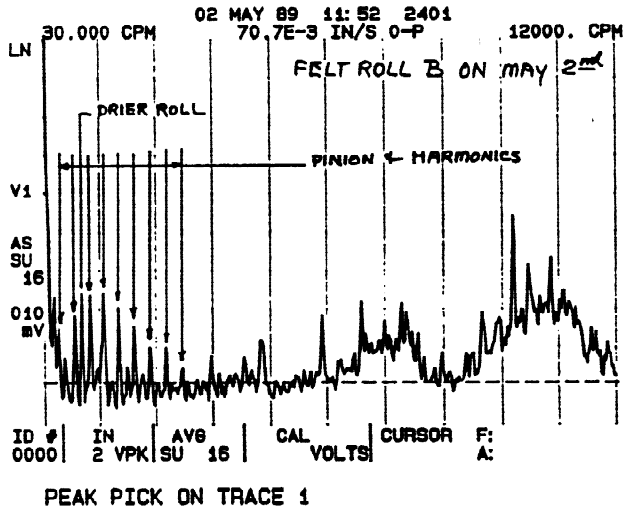


IN/S 0-P		CPM
28.5E-3	MOTOR RPM	1125.0
25.0E-3		5550.0
51.8E-3		14475.
25.0E-3		15150.
170.E-3	GMF	15449.

03-MAY-89 IH- HORIZONTAL AT POINT 1 ON MOTOR

16:05:45

FIGURE 11.10-D



IN/S 0-P	CPM
23.6E-3	240.00
17.8E-3	330.00
12.3E-3	450.00
20.0E-3	660.00
24.2E-3	810.00
23.9E-3	990.00
10.3E-3	1170.0
24.2E-3	1290.0
8.07E-3	1470.0
21.6E-3	1619.9
10.7E-3	1800.0
18.1E-3	1950.0
9.36E-3	2130.0
14.2E-3	2280.0
7.75E-3	2460.0

04-MAY-89 FELT ROLL B SHOWING PINION & HARMONICS

14:04:11

FIGURE 11.10-E

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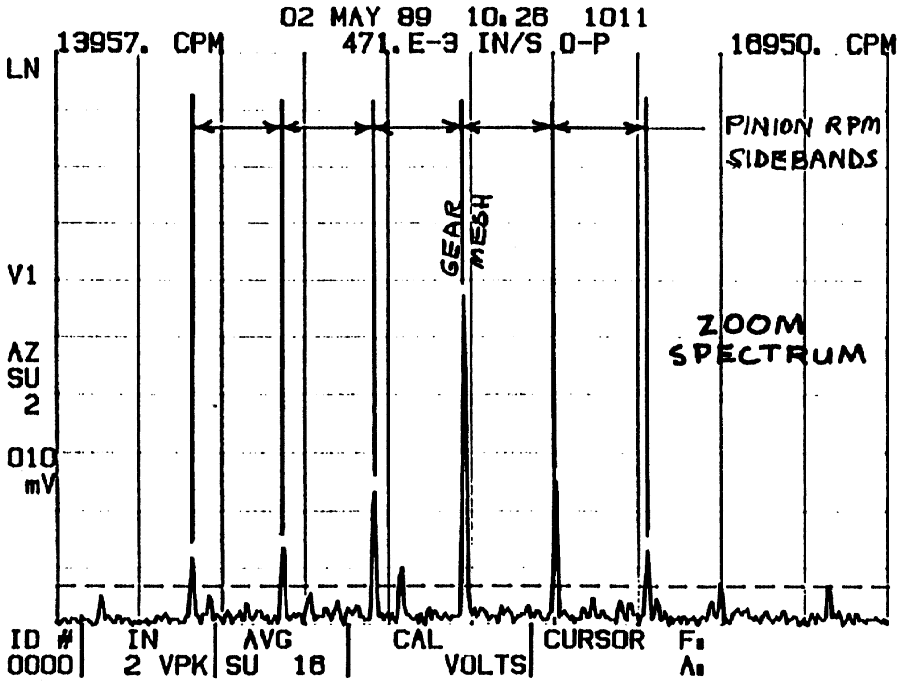


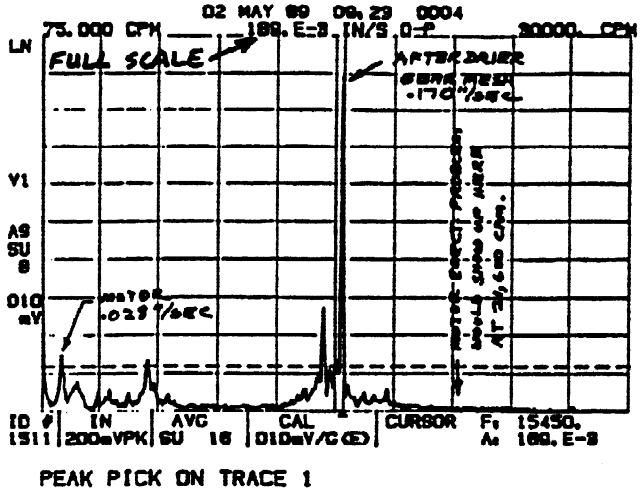
IN/S 0-P		CPM
55. 9E-3	-3x PINION RPM SIDEBAND	14444.
64. 5E-3	-2x PINION RPM SIDEBAND	14774.
110. E-3	-1x PINION RPM SIDEBAND	15104.
47. 3E-3		15202.
271. E-3	GMF	15427.
118. E-3	+1 x PINION RPM SIDEBAND	15757.
62. 4E-3	+2x PINION RPM SIDEBAND	16087.
34. 4E-3		16350.
32. 3E-3		16740.

03-MAY-89

IDLER PINION II

17: 46: 14

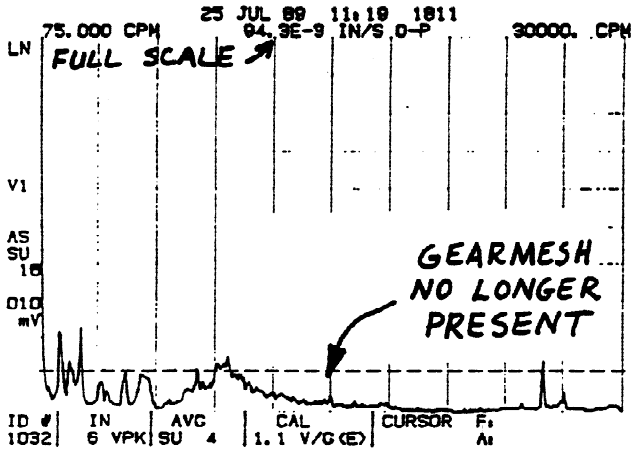
FIGURE 11.10-F



PEAK PICK ON TRACE 1

IN/S 0-P	CPM
28.5E-3	1125.0
25.0E-3	5550.0
51.8E-3	14475.
25.0E-3	15150.
170.E-3	15449.

BEFORE REPAIRS



PEAK PICK ON TRACE 1

IN/S 0-P	CPM
20.2E-3	1050.0
12.5E-3	1574.9
21.1E-3	2175.0
10.8E-3	8100.0
12.1E-3	9150.0
12.1E-3	9525.0
13.8E-3	9675.0
12.5E-3	26025.

AFTER REPAIRS

25-JUL-89

FIGURE 11.10-G

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CAUSE OF HIGH NOISE AND VIBRATION PROBLEMS ON A DRAW FRAME BEVEL GEAR SET

I. BACKGROUND

A high noise level on the Draw Frame 10 gearbox which drives the #7 roll was investigated on April 4, 1989. Both noise and vibration data were collected for analysis. Following is a report with: II. CONCLUSIONS, III. RECOMMENDATIONS, and IV. DISCUSSIONS.

II. CONCLUSIONS

A. A high noise level and moderate levels of vibration on the gearbox that drives #7 roll on Draw Frame 10 is caused by the input level gear set which has a looseness condition.

III. RECOMMENDATIONS

A. It is recommended that an overhaul of the gearbox be scheduled. Plan on changing the bearings associated with the input bevel gear set (pinion and gear) and also the bevel gear set itself.

IV. DISCUSSION

A. GENERAL

A very loud noise on the Flender gearbox, which is part of the drive system for Draw Frame 10 roll #7, was investigated on April 14, 1989. The gearbox is a Flender Model KZN 280 Serial Number 480.106.01 with a 14.44 to 1 ratio. The gearing information on Figure 1 should be filed for future reference since it was time consuming to obtain. The high speed gears are a bevel set with 16 pinion teeth and 26 gear teeth. The intermediate gears are a 17 tooth pinion and 47 tooth gear and the low speed set is a 14 tooth pinion and 45 tooth gear.

B. MECHANICAL FREQUENCIES

From vibration measurements it was determined that some variation in the motor speed is occurring. A speed of 1368 RPM was about average on the day measurements were made, therefore, it has been used as the motor speed.

MECHANICAL FREQUENCIES

Motor Speed---------- 1,368 CPM
Gear Mesh 1---------- 21,888 CPM
Intermediate Shaft Speed---------- 842 CPM
Gear Mesh 2---------- 14,311 CPM
Low Speed Shaft---------- 304 CPM
Gear Mesh 3---------- 4,263 CPM

The mechanical frequencies listed above have been used to perform an analysis, keeping in mind some variation in speed.

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C. VIBRATION

Many vibration spectra were taken on the gearbox, brake and motor. Only key spectral plots are used in the report but the other plots are available from our file. Plot 1 is a vibration spectrum from a horizontal measurement on the gearbox, showing .169 in/sec vibration at a frequency of 21,899 CPM. As seen in the mechanical frequency calculations and Figure 1, this frequency is from the input bevel gear set in the gearbox. Another reading, on the brake, is shown by Plot 2. Plot 2 is a zoom showing input bevel gear mesh frequency sidebanded by 1/2 pinion speed. These two plots indicate looseness in the high speed bevel gear set. This looseness causes peculiar loading of the gears, resulting in a very noisy gear mesh. The looseness could be between the pinion and its shaft or could be in the shaft itself if the bearings are worn. When an overhaul is planned, it may be necessary to have both the bearings and gears available since the exact cause is not entirely clear. What is clear is that the problem is in the input bevel gear set and that there is some looseness involved in the bevel pinion shaft.

Just plain wear of the bevel gear set could also cause the frequencies seen but the information available is that backlash was recently checked and was not found to be excessive.

D. NOISE

Noise spectra show clearly that the source of the loud noise is the high speed bevel gear set. In fact, the noise spectrum looks a lot like the vibration spectrum. Plot 3, taken at point A on Figure 1, shows the bevel gear mesh with sidebands. Note the extremely high noise level, 126 dB, at gear mesh frequency. This level is 100 times higher than the threshold requiring ear protection. Plot 4 is a zoom on gear mesh, taken at point A Figure 1, showing sidebands at 1/2 pinion running speed. The noise data agrees precisely with the vibration data but the sidebands at 1/2 running speed are much better defined.

An overhaul to correct the condition described earlier will eliminate the very high noise level at gear mesh frequency.

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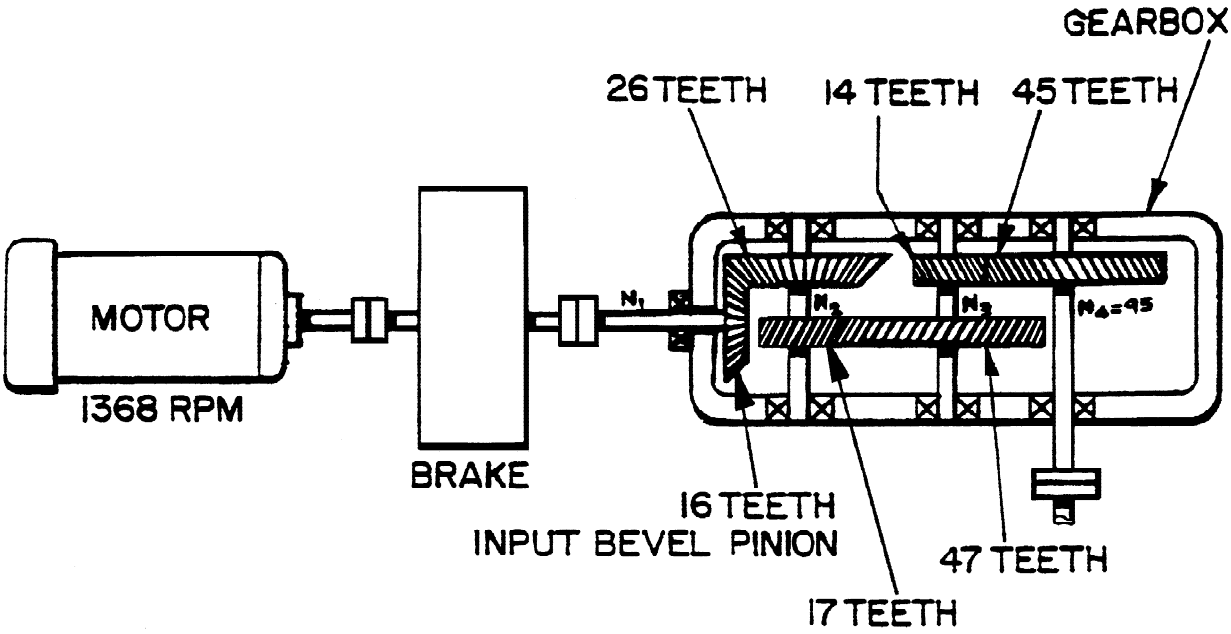
DRAW FRAME NO. 10 - ROLL NO. 7 DRIVE SYSTEM



N_1 = 1368 RPM
G_{m1} = 21,888

N_2 = 842
G_{m2} = 14,311

N_3 = 304
G_{m3} = 4,263

HIGH NOISE & VIBRATION FROM A BEVEL GEAR SET

FIGURE 1

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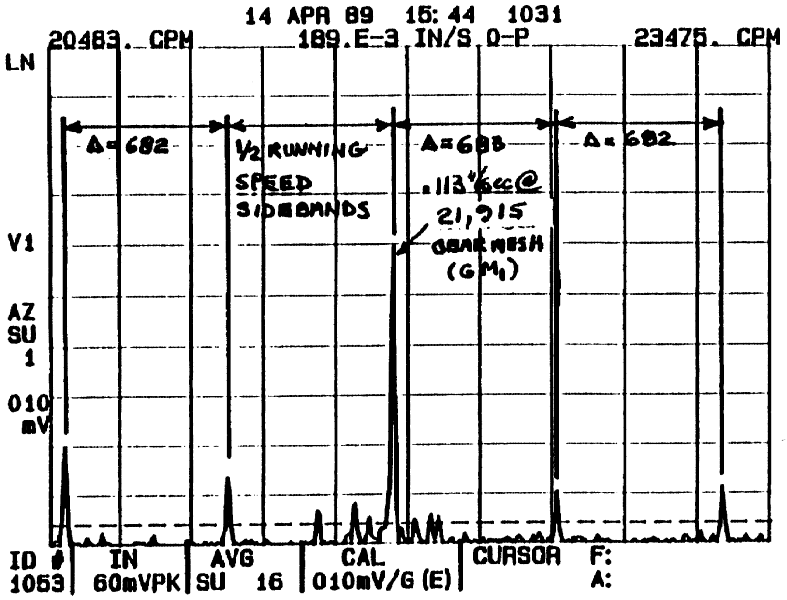

IN/S 0-P	CPM
38.0E-3	20543.
25.9E-3	21225.
12.9E-3	21600.
15.5E-3	21758.
10.4E-3	21818.
113.E-3	21915.
9.49E-3	22005.
11.2E-3	22073.
10.4E-3	22103.
19.8E-3	22598.
21.6E-3	23280.

17-APR-89 VIBRATION AXIAL ON BRAKE

15: 16: 06

PLOT 2

D FRAME 10

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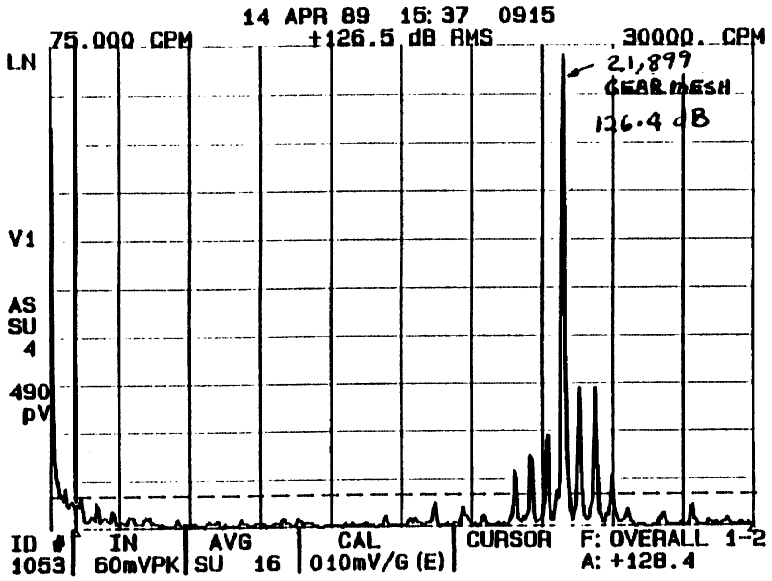

PEAK PICK ON TRACE 1

dB RMS	CPM
+105.3	675.00
+103.2	1350.0
+108.0	19800.
+110.1	20475.
+112.2	21224.
+104.3	21599.
+126.4	21899.
+115.8	22574.
+115.8	23249.
+107.3	23924.

17-APR-89 NOISE MEASUREMENTS AT GEARBOX

14: 43: 35

PLOT 3

D FRAME 10

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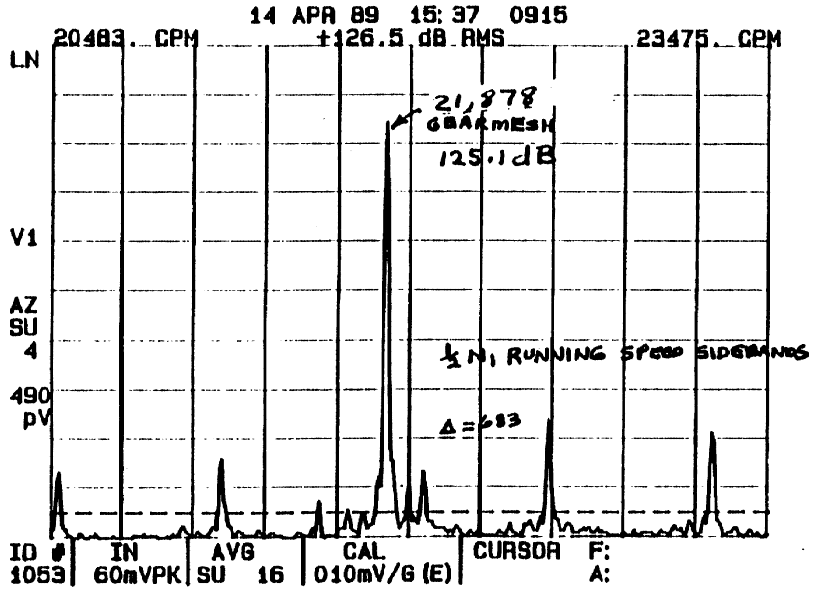


PEAK PICK ON TRACE 1

dB RMS	CPM
+108.9	20513. = $Gm_1 - N_1$
+110.6	21195. = $Gm_1 - \frac{N_1}{2}$
+103.8	21600.
+101.3	21720.
+100.5	21780.
+108.3	21848.
+125.1	21878 = Gm_1
+106.5	21975.
+108.9	22035.
+114.2	22560. = $Gm_1 + \frac{N_1}{2}$
+113.1	23243. = $Gm_1 + N_1$

17-APR-89

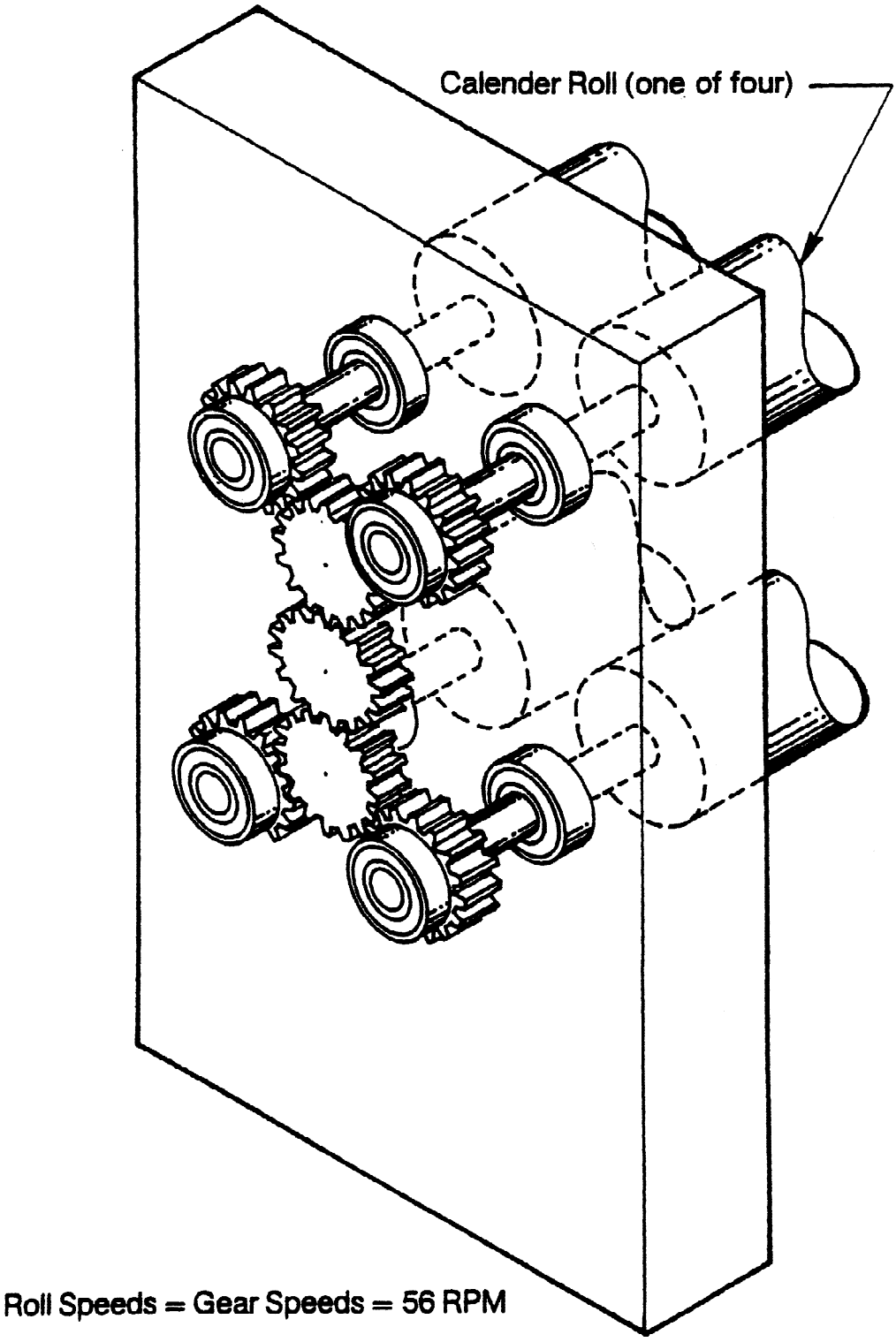
14: 44: 58

NOISE MEASUREMENT
ON GEARBOX
ZOOM ON GEARMESH + SIDEBANDS

PLOT 4

D FRAME 10

DETECTION OF CRACKED GEAR TEETH ON A 56 RPM CALENDER ROLL USING TIME WAVEFORM AND SPECTRAL ANALYSIS



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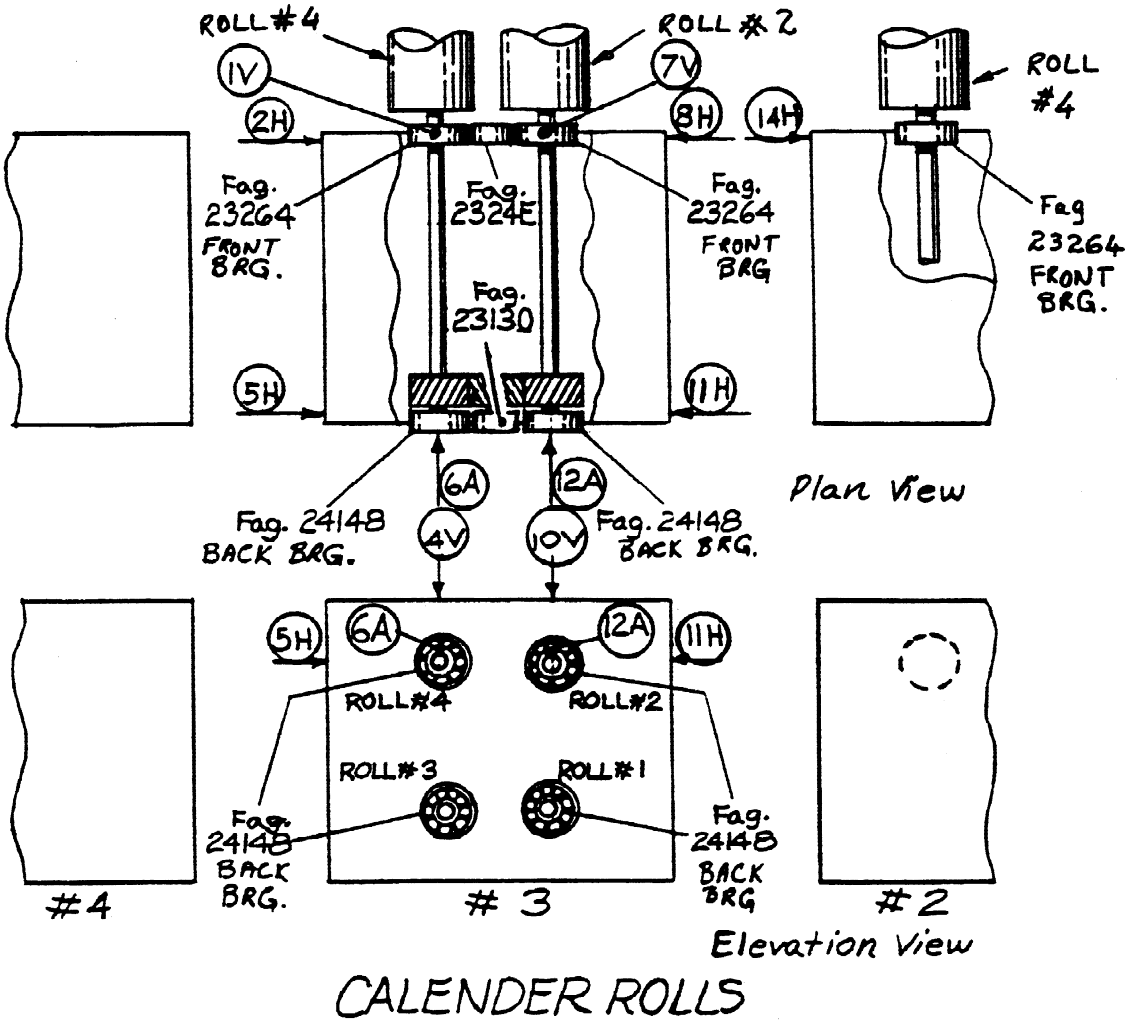

FIGURE 1
ACCELEROMETER LOCATIONS ON CALENDER ROLLS

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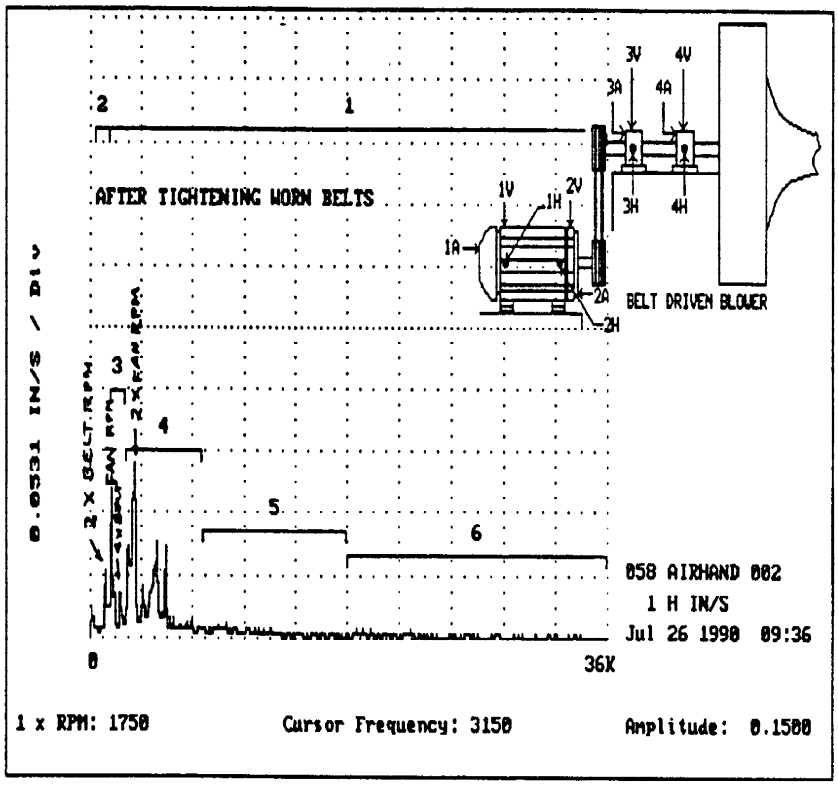
TABLE I. ANTICIPATED FREQUENCIES

	1F	2F	3F	4F	5F	6F	7F	8F	9F
Fag 23264									
FTF (1)	0.413	0.826	1.240	1.653	2.067	2.480	2.894	3.307	3.720
BPFI (1)	10.793	21.586	32.379	43.172	53.965	64.758	75.551	86.344	97.136
BPFO (1)	8.267	16.535	24.802	33.069	41.337	49.604	57.871	66.139	74.406
BSF (1)	3.428	6.856	10.284	13.712	17.140	20.568	23.996	27.424	30.852
Fag 24148									
FTF (2)	0.419	0.839	1.258	1.677	2.096	2.516	2.935	3.354	3.773
BPFI (2)	11.209	22.417	33.626	44.835	56.043	67.252	78.460	89.669	100.878
BPFO (2)	8.805	17.601	26.414	35.219	44.023	52.828	61.632	70.437	79.242
BSF (2)	3.773	7.547	11.320	15.094	18.868	22.641	26.414	30.188	33.961
Fag 2324E									
FTF (3)	0.387	0.774	1.162	1.549	1.936	2.323	2.710	3.097	3.483
BPFI (3)	7.356	14.712	22.068	29.424	36.780	44.136	51.492	58.849	66.205
BPFO (3)	5.033	10.066	15.099	20.132	25.165	30.199	35.232	40.265	45.298
BSF (3)	2.452	4.904	7.356	9.808	12.260	14.712	17.164	19.616	22.068
Fag 23130									
FTF (4)	0.424	0.847	1.272	1.695	2.119	2.543	2.967	3.391	3.814
BPFI (4)	12.701	25.401	38.102	50.803	63.504	76.204	88.905	101.606	114.306
BPFO (4)	10.172	20.343	30.515	40.686	50.858	61.029	71.201	81.373	91.544
BSF (4)	4.164	8.327	12.490	16.654	20.818	24.981	29.145	33.309	37.472
GMF	79.13	158.26	237.39	316.52	395.65	474.78	553.91	633.04	712.17
Roll Speed	0.953	1.906	2.859	3.812	4.765	5.718	6.671	7.624	8.577



2% x 0.500 = 0.0100 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
058 AIRHAND 002	1	H	IN/S	Jul 26 1990 09:36

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
90	0.051	0.0250	9540	5.451	0.0125
1080	0.617	0.0575 = 2X BELT			
1530	0.874	0.1250 = FAN RPM			
1800	1.029	0.0375 = MOTOR RPM			
2160	1.234	0.0375 = 4X BELT			
2700	1.543	0.0775 = 5X BELT (WAS .380 in/sec)			
3150	1.800	0.1500 = 2X FAN			
3690	2.109	0.0425			
4680	2.674	0.0800			
5040	2.880	0.0200			
5310	3.034	0.0750			
5580	3.189	0.0150			
5850	3.343	0.0150			
6390	3.651	0.0100			
6840	3.909	0.0125			
7200	4.114	0.0150			
9000	5.143	0.0100			

T - Threshold adjust Q - Quit

FIGURE 2

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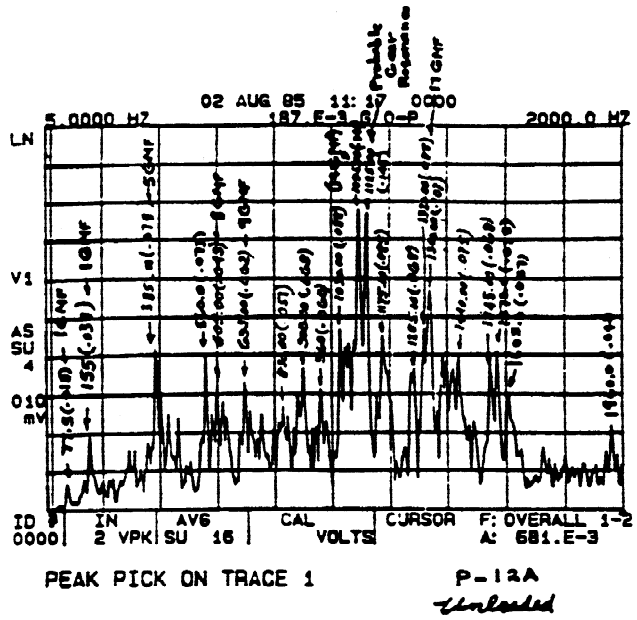


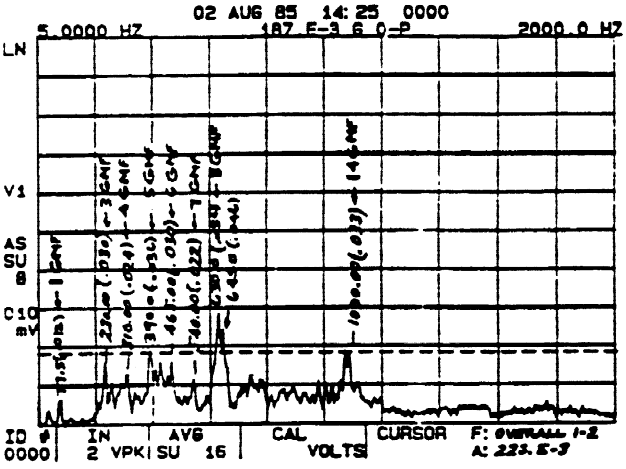
G 0-P	HERTZ
35.9E-3	155.00
77.7E-3	385.00
73.4E-3	560.00
61.5E-3	695.00
68.3E-3	900.00
146.E-3	1095.0
144.E-3	1125.0
107.E-3	1340.0
76.8E-3	1570.0

05-AUG-85

09: 06: 39

PLOT D
UNLOADED GEARS

CAL.ROLL#3



G 0-P	HERTZ
35.9E-3	390.00
53.8E-3	630.00
46.1E-3	645.00
35.0E-3	1075.0

05-AUG-85

12: 01: 00

PLOT E
LOADED GEARS

CAL.ROLL#3

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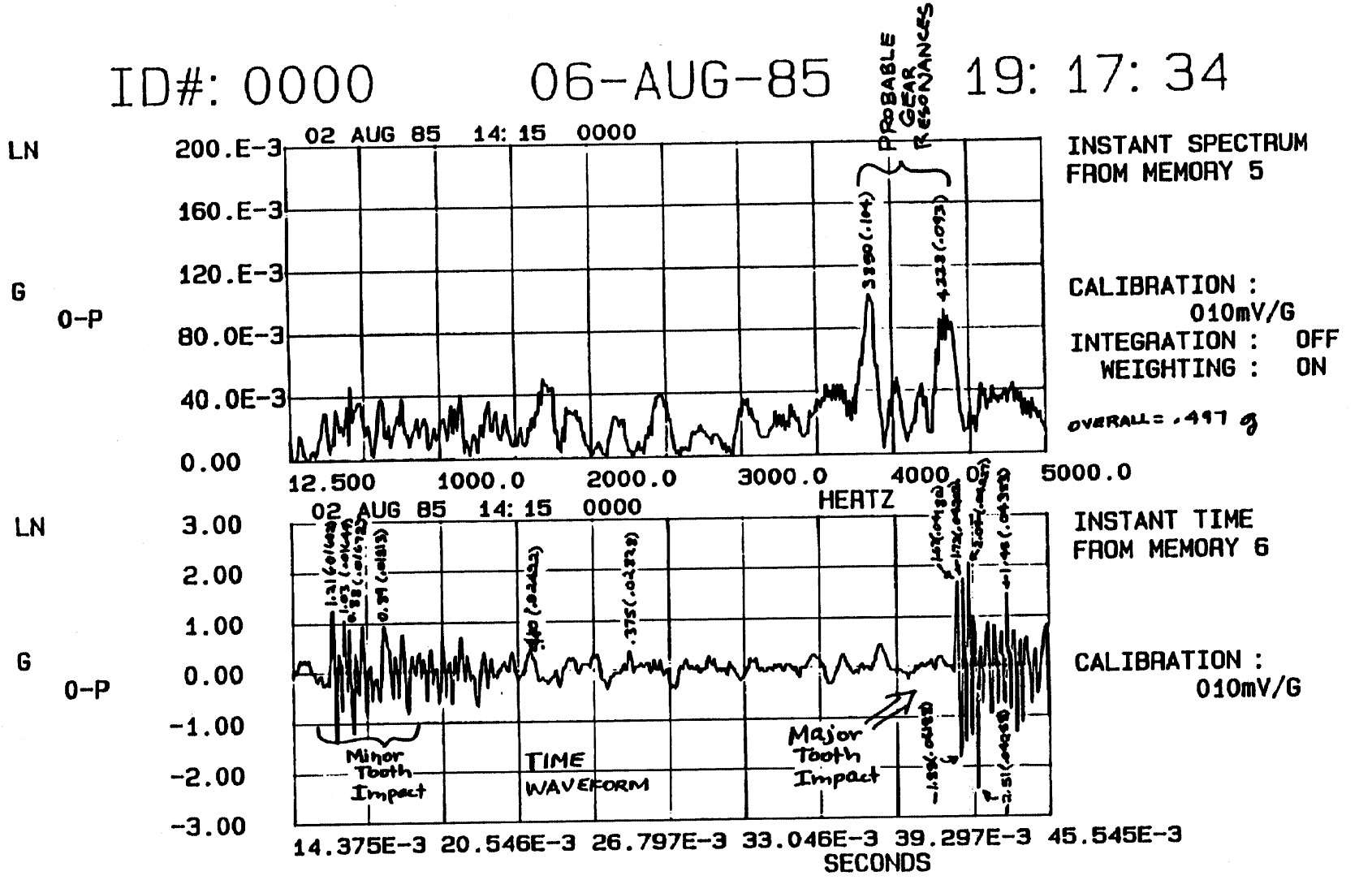
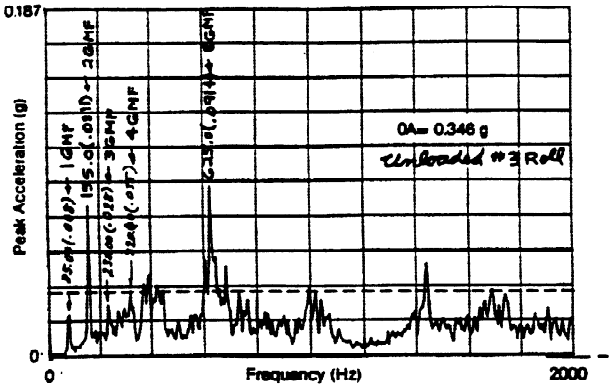


TOOTH IMPACT EVENTS

PLOT M
LOADED

CAL.ROLL#3



G 0-P	HERTZ
→ 81.1E-3	155.00
35.0E-3	320.00
39.3E-3	370.00
43.5E-3	390.00
38.4E-3	420.00
35.0E-3	440.00
51.2E-3	605.00
→ 91.4E-3	625.00
52.9E-3	645.00
35.9E-3	660.00
47.8E-3	685.00
34.2E-3	734.97
37.6E-3	994.97
34.2E-3	1020.0
49.5E-3	1440.0
35.0E-3	1685.0

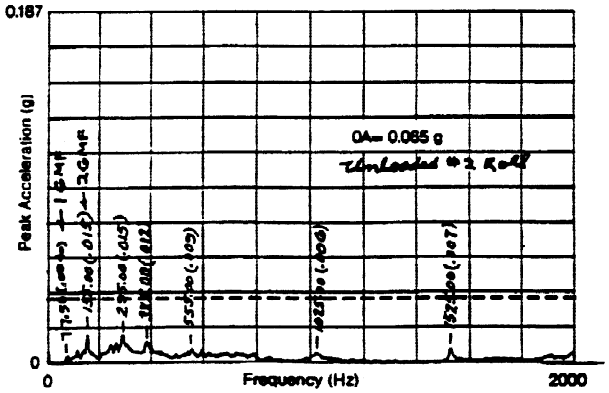
05-AUG-85

11: 04: 28

Calender #3
Position CA (axial)

FIGURE 3A

CAL. ROLL#3



G 0-P	HERTZ

Unloaded

05-AUG-85

11: 49: 46

Calender #2
Position CA (axial)

— COMPARISON OF 2 IDENTICAL ROLLS

FIGURE 3B

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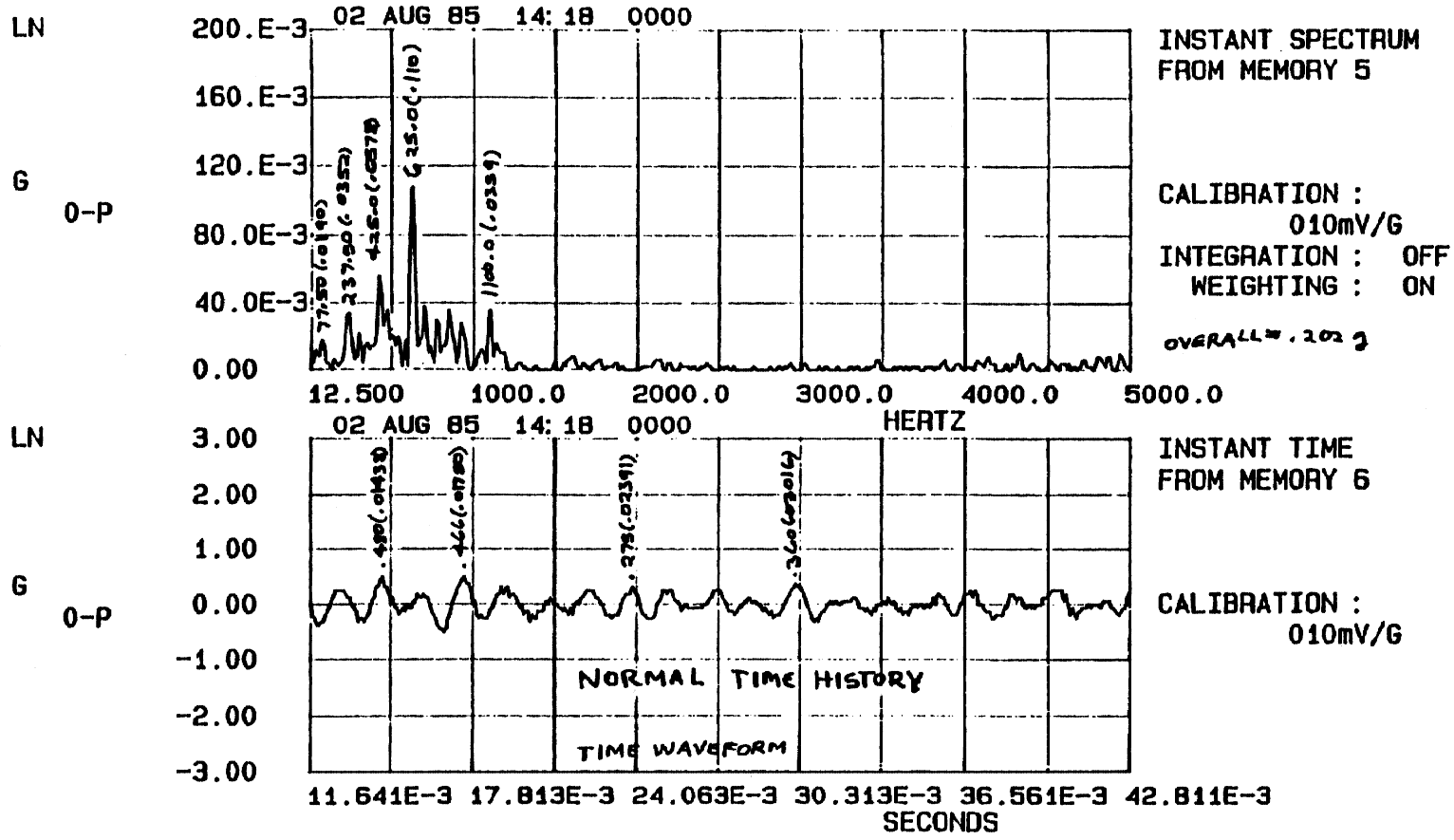
ID#: 0000 06-AUG-85 19: 23: 40



SAME GEAR HOUSING WHEN NOT SUBJECT TO IMPACTS

PLOT N
LOADED

CAL.ROL #3
POSITION 12A

HUNTING TOOTH VIBRATION PROBLEM ON A GEROTOR OIL PUMP LOCATED ON A CENTRIFUGAL AIR COMPRESSOR

Unusual Main Oil Pump Problem:

One of the more unusual and intriguing tests in our compressor analysis experience was conducted on a compressor found to have a problem with its main oil pump assembly. This pump was one of the positive displacement gerotor types mounted on the bullgear shaft, and had 13 tooth male and 14 tooth female gears (a sample gerotor mechanism is provided in Figure 18, but is not the unit described in this case history). During testing, Figure 16 shows that an excessive level of .535 in/sec was found at only 255 CPM on bullgear Position 3A (see Figure 1). Note that .535 in/sec at 255 CPM corresponds to an equivalent displacement of 40.0 mils! Subsequent zooming found that this subsynchronous frequency of 255 CPM was exactly one-seventh bullgear speed (1/7 F_{BG}). In addition, referring again to Figure 16, note the sidebands of 1/7 F_{BG} (255 CPM) spaced around bullgear running speed of 1785 RPM. Obviously, the next question asked was: "What significant event could be occurring at 1/7 bullgear speed - or, one event in each 7 revolutions of the bullgear shaft, and why did it have such an excessive amplitude?". Frankly, since the analyst had never before even seen this low frequency on a Centac™, he could hardly believe it was actually originating within this machine. However, the 255 CPM peak remained even when nearby machines were turned off. Still, for proof, the analyst had the operator cut off this compressor at which time the peak disappeared. Then, when the machine was started back up, the 255 CPM peak reappeared, and was at nearly an identical amplitude.

The answer to this perplexing problem finally began taking shape when an investigation of source frequencies from the main oil pump gearing was conducted. This showed the 255 CPM peak to be a harmonic of the so-called "hunting tooth frequency" - also referred to as the "tooth repeat frequency" as per Winterton (4). This has to do with how often a tooth on one gear will line up with a certain tooth on the mating gear. For example, referring to the Figure 18 sample gerotor mechanism, F_{HT} determines how often the #1 male tooth inside will line up with the #1 female tooth. The hunting tooth frequency is calculated as follows:

$$F_{HT} = \frac{(U_n)(GMF)}{(T_G)(T_P)} \qquad (3)$$

Where:

$\mathbf{F_{HT}}$ = Gear Hunting Tooth Frequency (CPM or Hz)

$\mathbf{U_n}$ = Number of unique assemblies for a given gear set (see below)

GMF = Gear Mesh Frequency (# T_G X F_{BG})

$\mathbf{T_G}$ = Number of teeth on gear

$\mathbf{T_P}$ = Number of teeth on pinion

$\mathbf{F_{BG}}$ = Bullgear RPM = Main Oil Pump RPM

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Note that U_n is numerically equal to the product of the prime factors that are common to the number of teeth on the gear and pinion. For example, if one had a 30-tooth gear and a 10-tooth pinion, U_n would equal to 10 (30 = 2 X 3 X 5; 10 = 2 X 5 thus; U_n = 2 X 5 = 10). For the main oil pump, the hunting tooth frequency (F_{HT}) was:

$$F_{HT} = \frac{(1)(13X1785)}{(13)(14)} = 127.5 \text{ CPM}$$

Thus, the 255 CPM peak was exactly $2F_{HT}$. And, therefore, the gerotor set was receiving a high amplitude pulse at twice the hunting tooth frequency. As a result, it was recommended that the plant access the main oil pump assembly and closely inspect it ASAP, looking for abnormalities.

When the plant inspected the pump, it found excessive wear of the key serving the main oil pump assembly. In fact, it had almost sheared, and was very loose within the keyway. Evidently, it was literally "bouncing" back and forth with each revolution of the drive shaft (thus, a possible explanation why the severe pulse occurred at twice hunting tooth frequency). In any case, Figure 17 shows the resultant spectrum after replacement of the oil pump key with a new properly sized one (there was some conjecture that an undersized key might have previously been installed). Note from Figure 17 that not only did the 255 CPM disappear, but the amplitude at bullgear frequency was notably reduced as well. In addition, sidebands at 1/7 F_{BG} were also eliminated around the bullgear speed peak.

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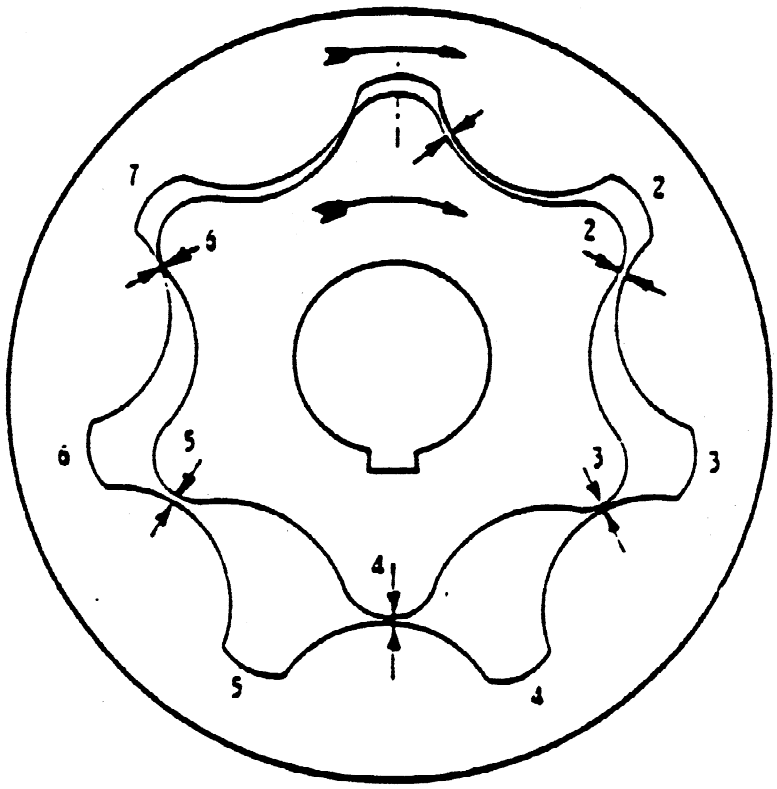

Courtesy Double A Products Co.

FIGURE 18
SCHEMATIC DIAGRAM OF THE GEROTOR MECHANISM

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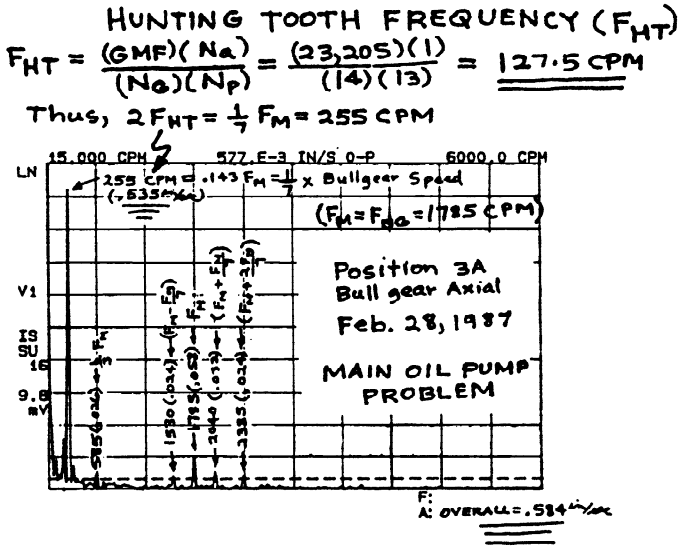
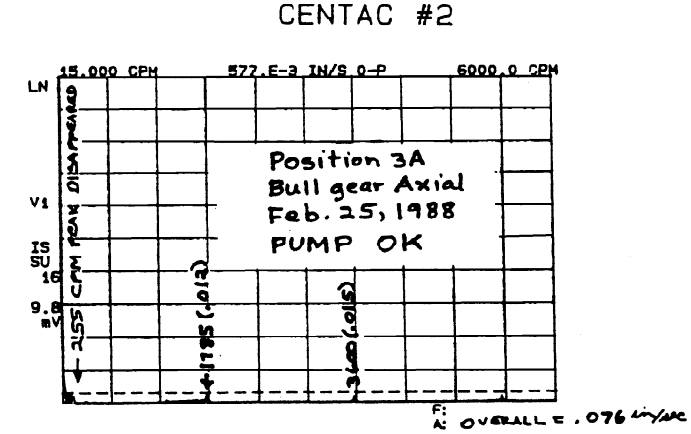


IN/S 0-P	CPM
52.7E-3	105.00
26.3E-3	150.00 ← $\frac{1}{12} F_M$
89.6E-3	195.00 ← $\frac{1}{9} F_M$
535.E-3 = 400 mils	255.00 ← $\frac{1}{7} F_M$
42.2E-3	315.00
18.4E-3	360.00 ← $\frac{1}{5} F_M$
26.3E-3	585.00 ← $\frac{1}{3} F_M$
23.7E-3	1529.9 - $(F_M - \frac{F_M}{7})$
58.0E-3	1785.0 - F_M
31.6E-3	2039.9 - $(F_M + \frac{F_M}{7})$
23.7E-3	2385.0 - $[F_M + 2\frac{F_M}{7}]$

FIGURE 16
EXCESSIVE VIBRATION AT GEROTOR HUNTING TOOTH 2ND HARMONIC



IN/S 0-P	CPM
18.4E-3	105.00

FIGURE 17
DISAPPEARANCE OF HUNTING TOOTH VIBRATION AFTER REPLACEMENT OF PUMP KEY

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DETECTION OF AN ELECTRICAL PROBLEM ON A PRIMARY AIR FAN MOTOR

I. BACKGROUND

A client with a Predictive Maintenance annual contract requested Technical Associates to perform a real-time analysis on all of their Primary and Secondary fans pointing out any problems in a brief report. All these units had the fan wheels integrally mounted on the motor shaft.

II. RESULTS AND CONCLUSIONS

The H-1 Primary Fan pictured in Figure 1 was found to have some problems. Both horizontal and axial spectra from the outboard and inboard motor bearings were dominated by 1X RPM vibration as demonstrated by Figures 2 and 3 taken at Positions 1A and 1H which had amplitudes of .320 in/sec and .375 in/sec, respectively. Both of these amplitudes exceeded the general alarm level of .275 in/sec for integrally shaft-mounted fans.

Table I compares the amplitude and phase at 1X RPM at each measurement location. Note that the horizontal phase difference of 14° (between outboard and inboard bearings) compared closely with the vertical phase difference of 29°. This indicated the most probable cause of vibration to be unbalance. Since an integral fan is also an overhung fan, the axial measurement is also affected by the imbalance.

Figure 4 is a zoom spectrum taken at Position 1A for the purpose of separating 2X RPM from 2X line frequency (7200 CPM). Note that 7200 CPM vibration was pulsating and the motor had an audible pulsing sound which is often indicative of an electrical problem. This zoom spectrum revealed a 2X electrical line frequency (7200 CPM) amplitude of .124 in/sec with pole pass frequency (F_P = Slip Frequency X # Poles) sidebands around 7200 CPM. A nominal 3600 RPM motor has 2 poles and the slip frequency was 38 CPM. This indicated a variable gap between the rotor and stator (which might have been effected by the unbalance of the fan wheel).

III. RECOMMENDATIONS

A. It was recommended to first balance the fan wheel. Because the phase analysis results shown in Table I indicated predominant static unbalance (only 15° to 30° difference in radial phase between the outboard and inboard bearings), it was felt only a single-plane balance would be required.

B. After completing the fan wheel balancing, it was recommended that measurements at all locations shown in Figure 1 be repeated. The primary interest here would be to determine if the high level of vibration at 2X electrical line frequency (.124 in/sec at $2F_L$) shown in Figure 4 at Position 1A would now be reduced to acceptable levels (hopefully below .06 in/sec at $2F_L$). If not, it was pointed out the plant would have to closely trend vibration at this frequency in future surveys.

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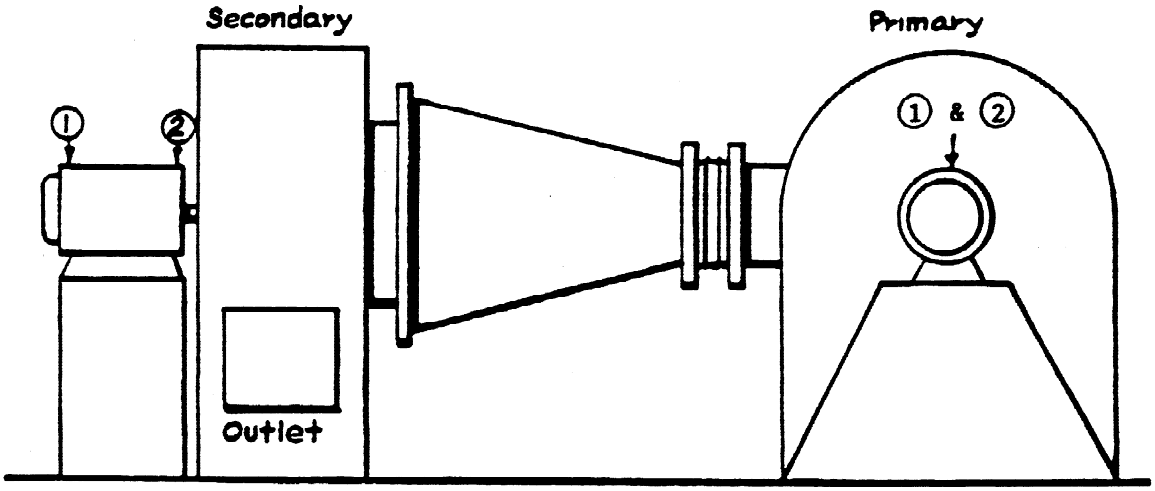

FIGURE 1
PRIMARY AND SECONDARY AIR FANS

POSITION NO.	1H	2H	1V	2V	1A	2A
VELOCITY (in/s) PHASE (Deg.)	.320 90°	.346 104°	.274 205°	.259 176°	.375 336°	.338 316°
PHASE DIFFER.	1H-2H=14°		1V-2V=29°		1A-2A=20°	

TABLE I
COMPARISON OF AMPLITUDE & PHASE AT 1X RPM AT EACH MEASUREMENT LOCATION

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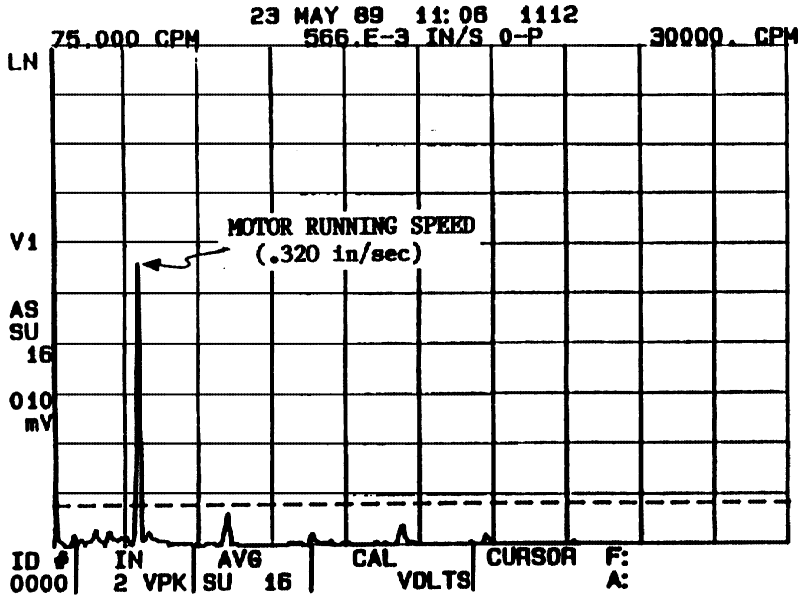

IN/S 0-P	CPM
320 E-3	3525.0

23-MAY-89

14: 39: 00

OVERALL = .33 "/SEC

1 HORIZONTAL

PLOT 3

FIGURE 2

H 1 PRI

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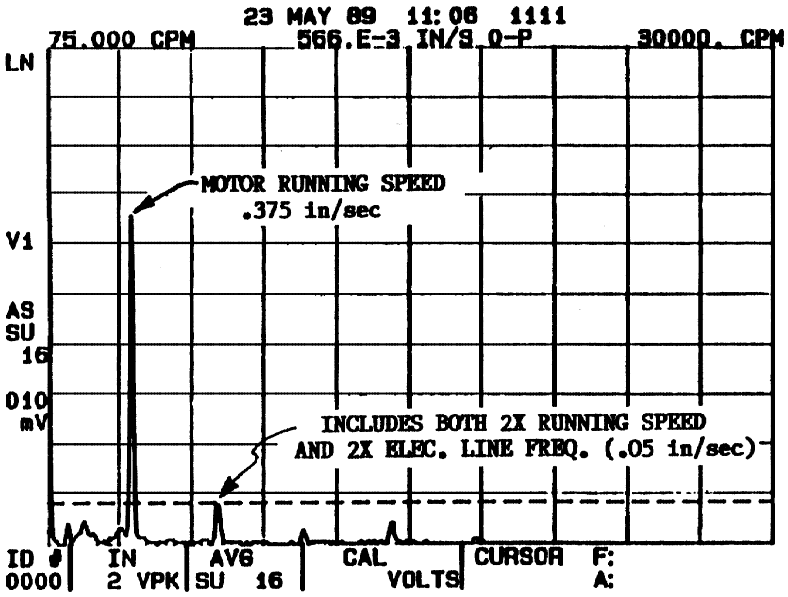

PEAK PICK ON TRACE 1

POSITION 1A

IN/S 0-P	CPM
375.E-3	3525.0

23-MAY-89

OVERALL = .377 "/SEC

AXIAL

14: 37: 59

PLOT 1

FIGURE 3

H 1 PRT

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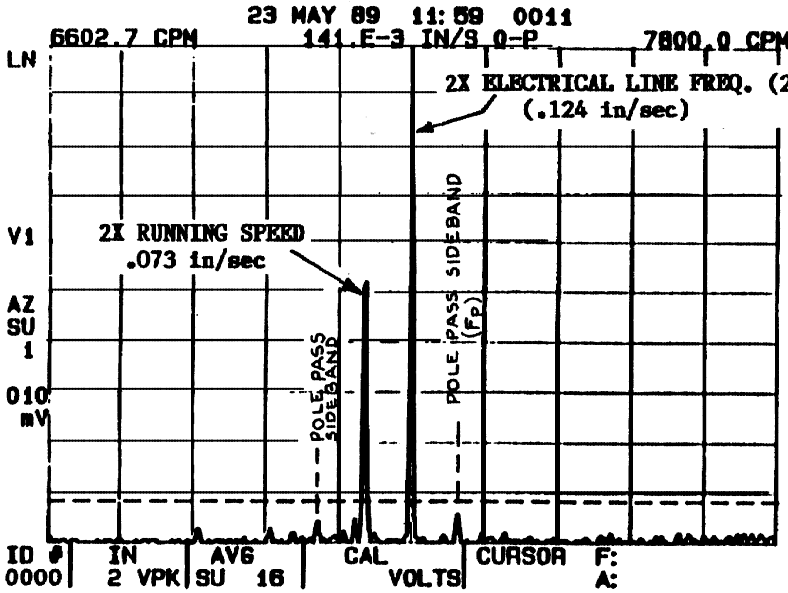

PEAK PICK ON TRACE 1

POSITION 1A

IN/S 0-P	CPM
73.4E-3	7124.7
124.E-3	7200.0

23-MAY-89 ZOOM SPECTRUM AROUND 7200 CPM AREA

14: 47: 32

PLOT 2

FIGURE 4

H 1 PRI

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SUMMARY OF RESULTS AND RECOMMENDATIONS FOR MACHINES FOUND TO HAVE PROBLEMS

(SORTED FIRST BY PROBLEM PRIORITY, THEN BY PROBLEM SEVERITY WITHIN PRIORITY) AS OF: 5/23/89

RECOMM. WORK ORDER	MACHINE COMPUTER NAME	INDICATED PROBLEM(S): UNBALANCE	MISALIGNMENT/SOFT FOOT	BENT SHAFT	LUBRICATION PROBLEM	BEARING PROBLEM	MECHAN. LOOSENESS	BELT/SHEAVE PROBLEM	RESONANCE PROBLEM	GEAR PROBLEM	BLADE PASSING PROB.	CAVITATION	MOTOR ELECTRICAL	OIL WHIRL/WHIP	LOOSE FOUNDATION	VIB. MONITOR PROB.	SEAL/ROTOR RUB	OTHER (SEE REMARKS)	PRIORITY OF PROBLEM	HIGHEST OVERALL VIBRATION: POSITION NO.	LEVEL: m = mils, i = in/sec, a = g's	RECOMMENDATIONS & REMARKS	SEE PLOT NO.
#6	H-5 PRIMARY FAN	●				●													2	1H 2V	.483 .307	Unbalance shown in Plot 18A by .470 in/sec at 1 x RPM. Incipient bearing problem indicated by Plots 18B & 18C by peak at 3.078 X RPM with level of .037 in/sec, but need bearing model number to confirm. Recommend balancing fan wheel. Trend possible deterioration of bearing frequency after confirming bearing model number.	18A 18B 18C 18D
#7	H-2 SECONDARY FAN	●					●												2	2H	.409	Fan wheel unbalance is indicated by Plot 10A while mechanical looseness is evidenced by Plot 10B. Recommend balancing fan. This may reduce the looseness problem which is not a problem cause, but a reaction to other problems imposed.	10A 10B
#8	SECONDARY FAN H586												●						2	2H 1V	.160 .254	Electrical problem indicated by .140 in/sec at 7200 CPM at position 2H in Plot 24. Recommend trending this high level of electromagnetic vibration very closely in future surveys. Such electrical vibration levels can cause damage to armature windings and may induce possible rub between rotor and stator. Not necessary for repair at this time. However, a close inspection of rotor bars, windings, etc. should be made.	24
—	H-1 PRIMARY FAN	●											●						3	1A	.3751	Fan wheel unbalance indicated (Plots 1 & 3) plus a motor electrical problem (Plot 2). Electrical problem likely caused by air gap eccentricity. Recommend trending possible deterioration of these problems in future vibration surveys.	1 2 3
—	H-6 PRIMARY FAN	●																	3	1A	.375	Marginal unbalance with level of .372 in/sec at 1A shown in Plot 20. Trend in future surveys.	20
—	H-6 SECONDARY FAN	●																	3	1A	.346	Marginal unbalance with level of .339 in/sec at 1A shown in Plot 21. Trend in future surveys.	21
—	SPARE FAN H3 AND 4	●																	3	1A	.340	Marginal unbalance with level of .328 in/sec at 1x at 1A shown in Plot 22. Trend in future surveys.	22

*① = TAKE IMMEDIATE REPAIR ACTION DUE TO SEVERE PROBLEM(S) NOTED (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
2 = SCHEDULE REPAIRS TO CORRECT PROBLEMS NOTED AT CONVENIENCE (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
3 = DESIGNATED PROBLEM DOES NOT WARRANT ACTION AT THIS TIME (CONTINUE ONLY WITH RELIABILITY SURVEY TREND ANALYSIS)

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TRENDING OF A MOTOR ELECTRICAL PROBLEM WHICH EVENTUALLY CONCLUDED WITH CATASTROPHIC FAILURE

I. BACKGROUND

A customer's air handler motor that Technical Associates monitored bimonthly in a Predictive Maintenance vibration program had a fairly high, but stable amplitude at 7200 CPM (2X electrical line frequency) as seen in the Figure 1 spectra dated September 26, 1988 and January 31, 1989 with 7200 CPM amplitudes of .095 in/sec. Since this and other spectra taken at other vibration is indicative of either an uneven stationary air gap due to rotor eccentricity or shorted stator laminations. During the March 22, 1989 survey the 7200 CPM amplitude increased to .265 in/sec along with an increase to .130 in/sec from .095 in/sec at 1800 CPM (1X motor RPM). The increase at 1800 CPM was possibly due to the localized heating of the stator which accompanies shorted rotor laminations. This localized heating can result in an eccentric rotor and bent motor shaft, imbalance, lubricant breakdown and eventual lockup. On the next survey, May 23, 1989, the 7200 CPM and 1800 CPM amplitudes had both increased to levels of .265 in/sec and .2025 in/sec respectively as seen in Figure 1. The 7200 CPM frequency also had harmonics of 7200 CPM at 14,400, 21,600 and 28,800 CPM.

II. CONCLUSIONS

A. The motor had worsening electrical condition.

III. RECOMMENDATIONS

A. The client was strongly recommended on March 22 to replace/overhaul the motor before catastrophic failure due both to the excessive amplitude of .238 in/sec at 7200 CPM and the fact it had so decisively trended up from .095 in/sec at 7200 CPM during the previous survey. When no repair actions were taken and when this frequency continued increasing up to .265 in/sec at 7200 CPM on May 23, Technical Associates left two written "Immediate Action Required" reports with each of two people before leaving the premises. Once again, repair action was urgently requested.

IV. RESULTS

The motor seized approximately 2 weeks after the May 23 survey when no attempts to replace/overhaul the unit were made. It was reported there was considerable damage to the motor since its rotor contacted the stator, possibly due to localized, unequal heating of the rotor.

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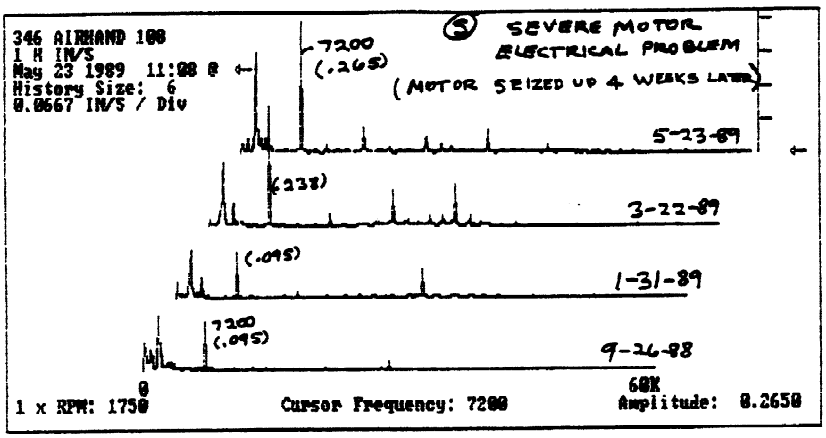

006% x 0.500 = 0.0300 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
346 AIRHAND 108	1	H	IN/S	May 23 1989 11:0

Frequency	1 x RPM	Amplitude
900	0.51	0.0300
1800	1.03	0.2025
7200	4.11	0.2650
14400	8.23	0.0550
21600	12.34	0.0350
28800	16.46	0.0475

6% x 0.500 = 0.0300 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
346 AIRHAND 108	1	H	IN/S	Mar 22 1989 09:5

Frequency	1 x RPM	Amplitude
1800	1.03	0.1300
3000	1.71	0.0400
7200	4.11	0.2375
21600	12.34	0.0675
28800	16.46	0.0775

6% x 0.500 = 0.0300 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
346 AIRHAND 108	1	H	IN/S	Jan 31 1989 14:3

Frequency	1 x RPM	Amplitude
150	0.09	0.0325
1800	1.03	0.0950
3000	1.71	0.0425
7200	4.11	0.0925
28800	16.46	0.0575

6% x 0.500 = 0.0300 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
346 AIRHAND 108	1	H	IN/S	Sep 26 1988 16:3

Frequency	1 x RPM	Amplitude
150	0.09	0.0525
900	0.51	0.0425
1200	0.69	0.0300
1800	1.03	0.1075
7200	4.11	0.0950

TRENDING OF A MOTOR ELECTRICAL PROBLEM WHICH EVENTUALLY CONCLUDED WITH CATASTROPHIC FAILURE

FIGURE 1

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REJECTION OF A CENTRIFUGAL CHILLER MOTOR DURING ACCEPTANCE TESTING DUE TO NOTICEABLE ELECTRICAL PROBLEMS

I. BACKGROUND

Technical Associates was asked to provide expertise in acceptance testing of a new York Chiller with a Siemens-Allis flange mounted motor as seen in the machine drawings of Figures 1 and 2. If problems were found, recommended actions to resolve the problems were requested.

II. RESULTS

Upon startup, a large nominal 7200 CPM vibration was noted at .139 in/sec amplitude as seen in the upper instantaneous spectrum of Figure 3. It needed to be determined if the so-called 7200 was caused by 2X electrical line frequency (2X 3600 CPM) due to an electrical problem, or if it was caused by 2X motor speed possibly due to a looseness or alignment problem. Since the motor rotated at 3591 RPM as seen in the "zoom" spectrum of Figure 4, and the maximum frequency was 12,000 CPM, the spectral resolution of 30 CPM did not allow one to determine which of the above cases existed. To resolve the situation, synchronous time averaging was employed.

Synchronous time averaging uses a synchronizing signal (photocell, or electromagnetic pickup) to sample the vibration in a manner that filters out nonsynchronous vibration such as random background vibration and passes only synchronous vibrations from the machine being analyzed.

The lower spectrum of Figure 3 is synchronous time averaged. In this spectrum (lower spectrum of Figure 3) note the so-called "7200 CPM amplitude" is only .011 in/sec compared to the instantaneous spectrum amplitude of .139 in/sec at this frequency in the upper spectrum. This indicated only approximately .011 in/sec was present at 2X motor speed while actual 7200 CPM vibration at 2X line frequency was almost .130 in/sec which was unacceptably high for a new motor installation (note that vibration at 7200 CPM would not be synchronous to a motor speed of 3519 RPM, and therefore would be "filtered out" from a synchronous time averaged spectrum if a sufficient number of averages are taken). To further verify this conclusion, a coastdown test was collected as shown in Figure 5. Spectra captured just prior to shutting off the electricity show a dominant 1X motor vibration (3600 nominal CPM) and another dominant vibration at 7200 CPM. As soon as power to the motor was disconnected, note that the 7200 CPM peak almost instantaneously disappeared as an electrically induced vibration would. This was further evidence the problem was electrical. Further examination of other spectra found the electrical problem was due to stator eccentricity or shorted stator laminations.

II. CONCLUSIONS

A. An electrical problem due to stator eccentricity or shorted stator laminations existed with this new motor. If accepted, it would likely experience a noticeably shortened operating life.

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III. RECOMMENDATIONS

A. Reject this new motor and take actions either to rectify its problems or replace it with another motor not having such problems.

B. Perform an acceptance test on the repaired or replacement motor.

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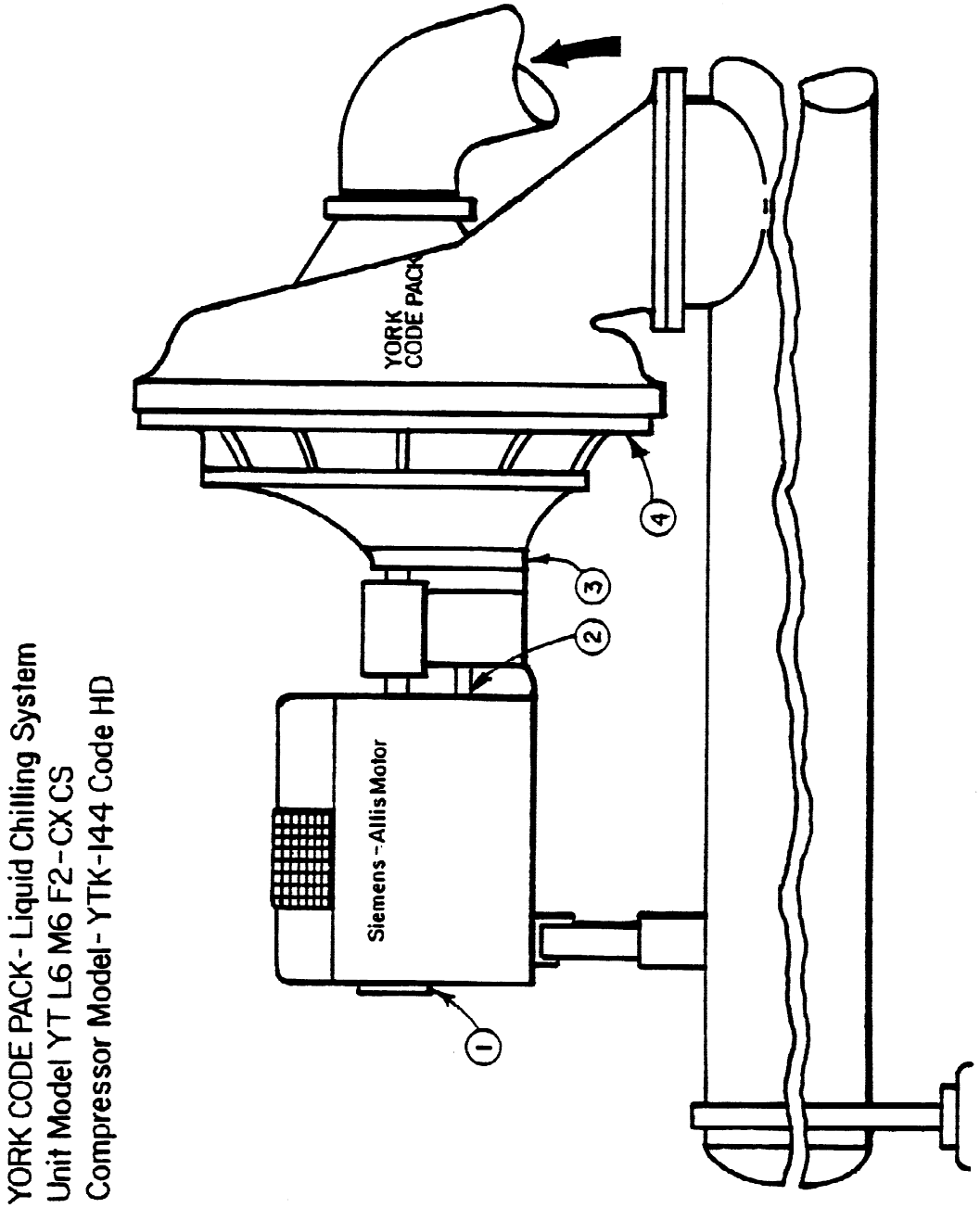
REJECTION OF A CENTRIFUGAL CHILLER MOTOR DURING ACCEPTANCE TESTING
DUE TO NOTICEABLE ELECTRICAL PROBLEMS



YORK CODE PACK - Liquid Chilling System
Unit Model YT L6 M6 F2-CX CS
Compressor Model - YTK-144 Code HD

FIGURE 1
Vibration Data Collection Points On York Code Pak Chillers

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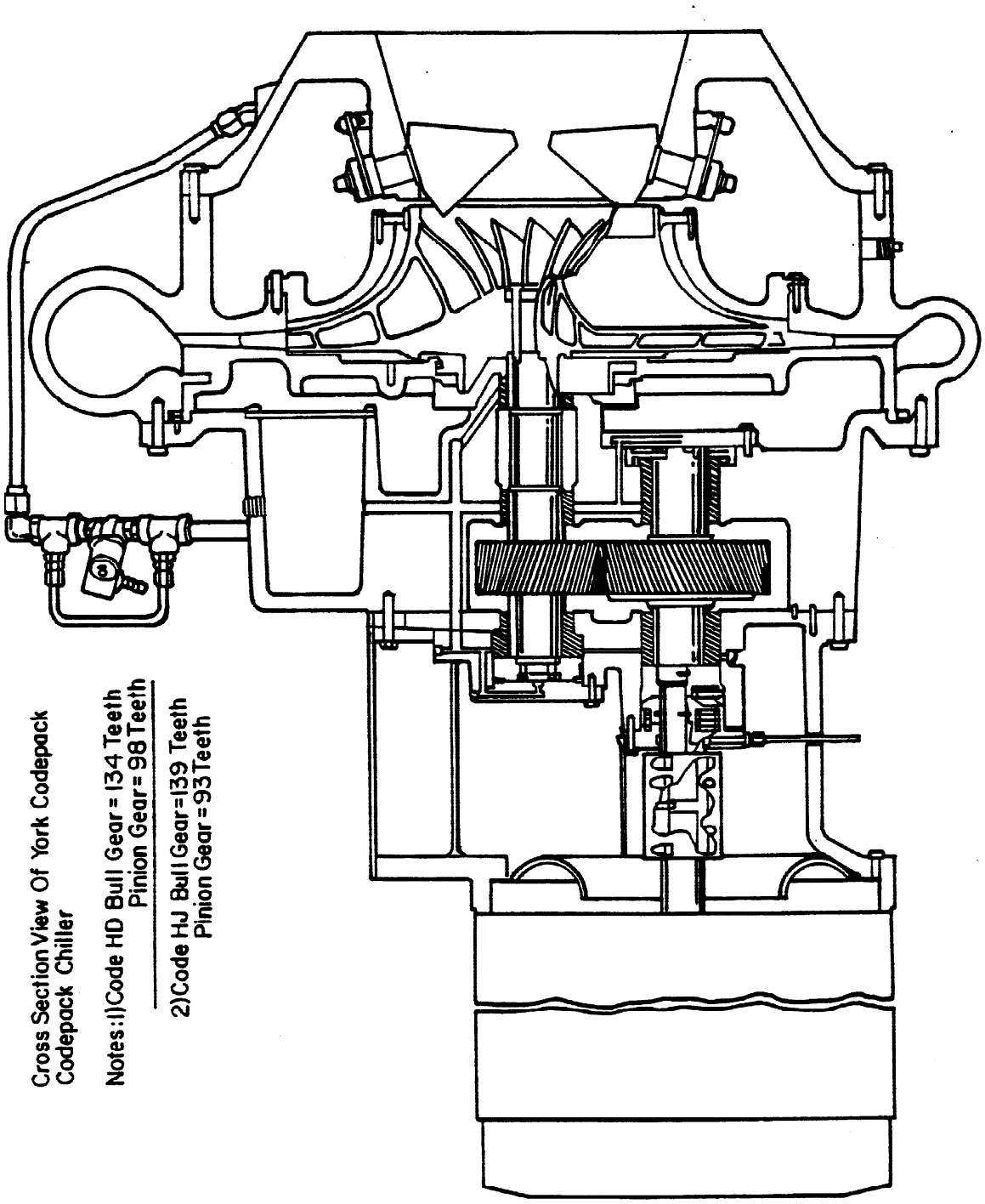
Cross Section View Of York Codepack
Codepack Chiller
Notes: 1) Code HD Bull Gear = 134 Teeth
Pinion Gear = 98 Teeth
2) Code HJ Bull Gear = 139 Teeth
Pinion Gear = 93 Teeth

FIGURE 2

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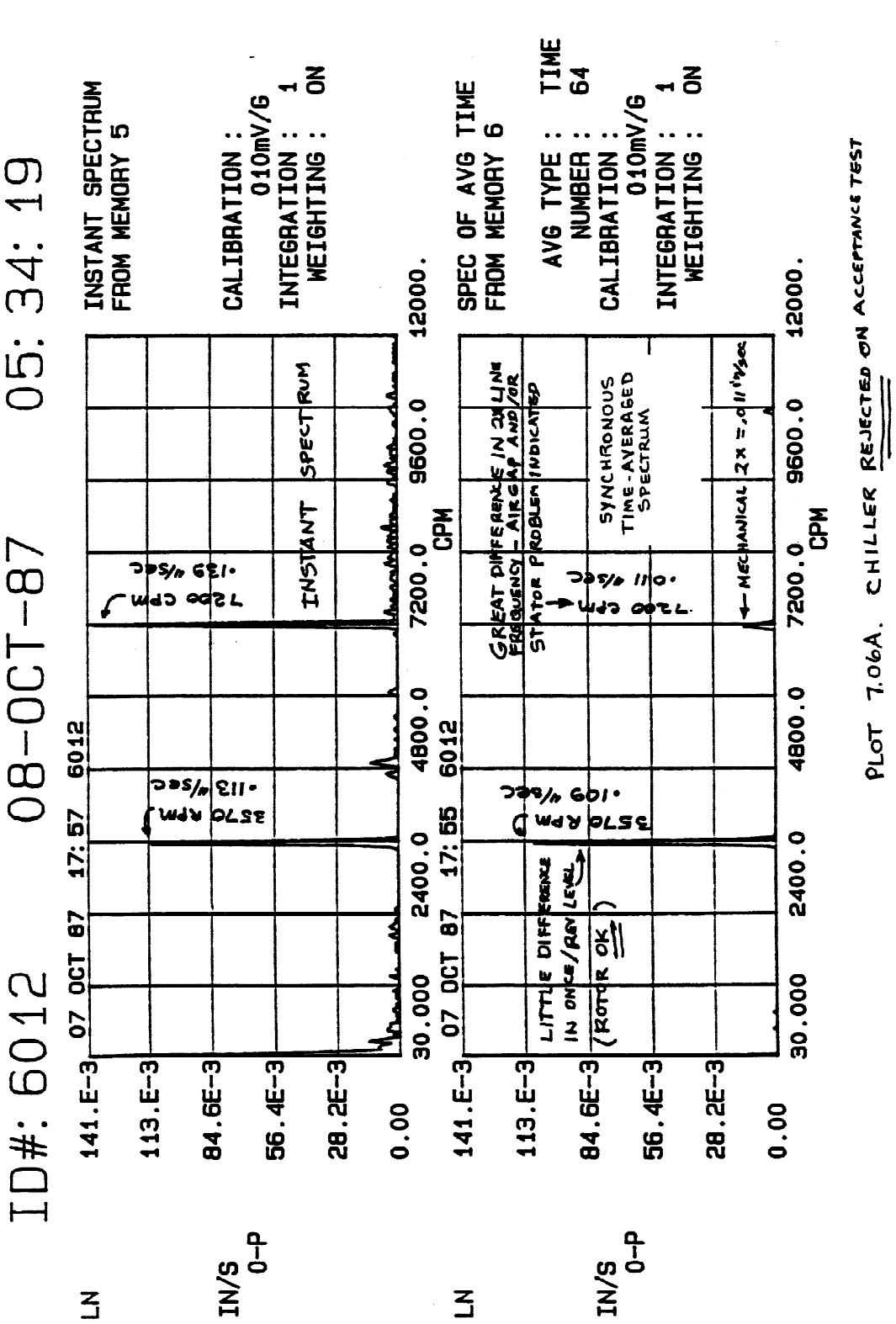

PLOT 7.06A. CHILLER REJECTED ON ACCEPTANCE TEST

PLOT 16

FIGURE 3

CHILLER # 6

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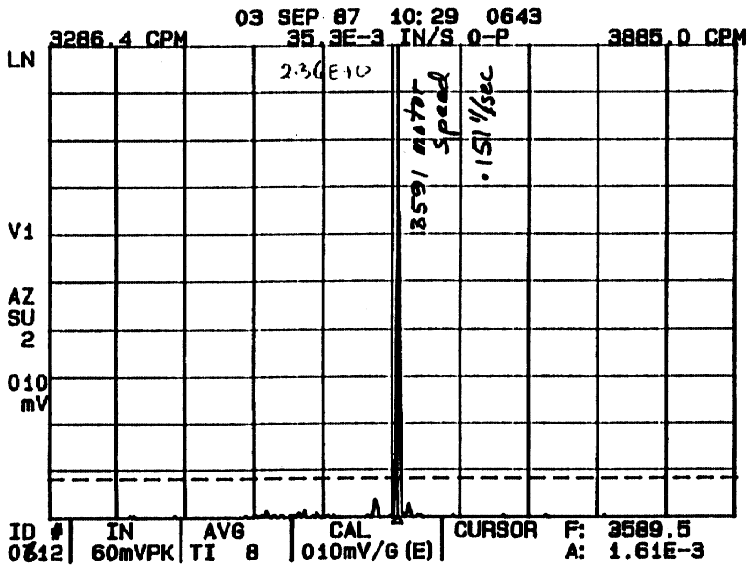

PEAK PICK ON TRACE 1

IN/S 0-P	CPM
22.9E-3	3591.0
151.0E-3	

01-JAN-87

08:51:08

CHILLER #6

ZOOM ON MOTOR SPEED

TESTS

PLOT 13

FIGURE 4

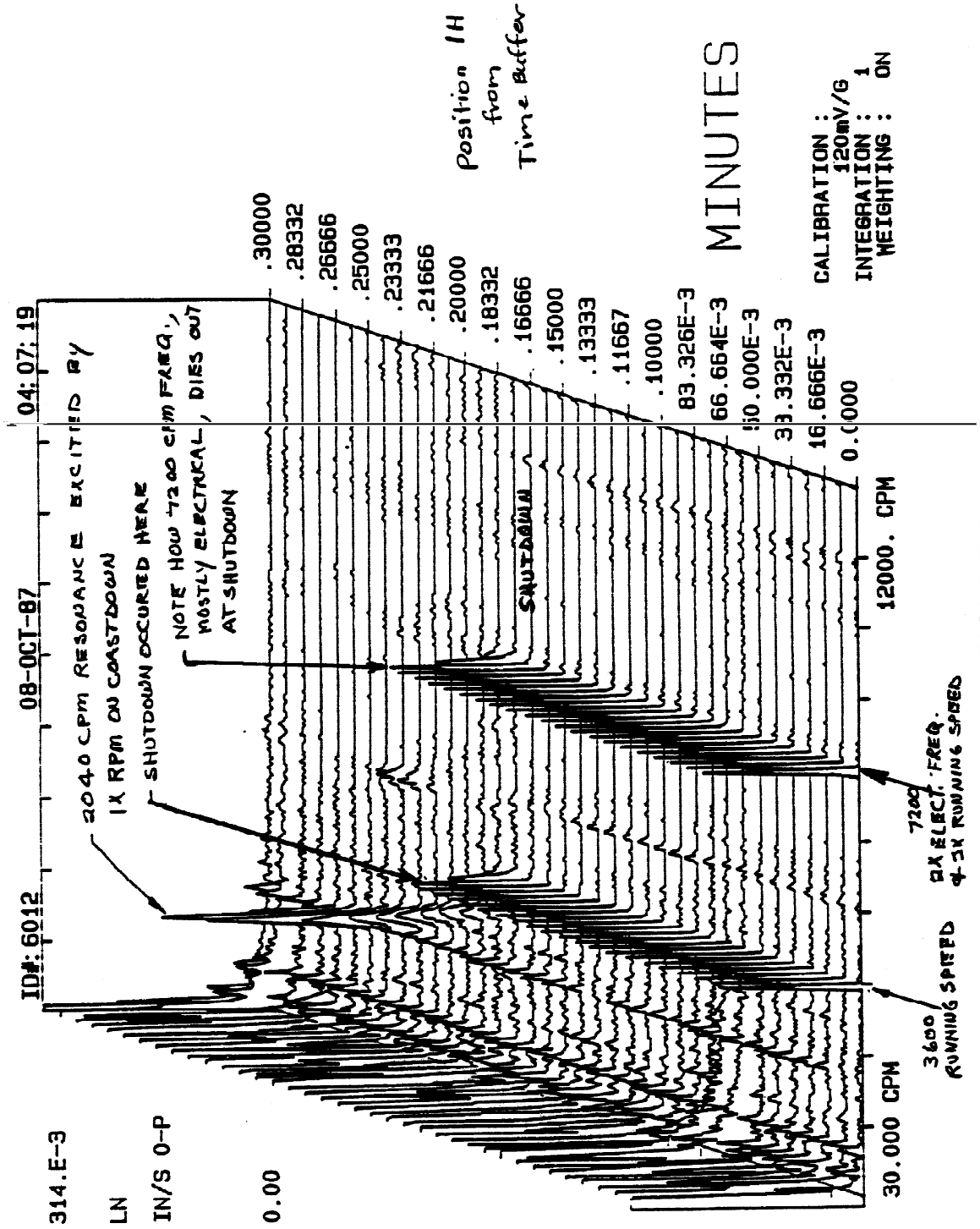

FIGURE 5

© Copyright 1997 Technical Associates of Charlotte, P.C.

DETECTION OF LOOSE ROTOR BARS IN A DRYER EXHAUST FAN MOTOR

I. BACKGROUND

An annual contract was awarded to Technical Associates to perform quarterly Real Time Analyzer (RTA) vibration measurements on 7 high starting torque electric motors for the specific purpose of evaluating motor electrical conditions. These RTA measurements were in addition to the normal monthly PMP measurements already being performed. These measurements were to include incremental "snapshots" of instantaneous vibration spectra during machine run-up (0 - 300,000 CPM range), peak-hold spectra acquired during run-up (0 - 300,000 CPM range) and zoom spectrum analysis of running speed (F_M), line frequency (F_L), two times line frequency ($2F_L$) and RBPF (with multiples) during normal operation. Figure 1 shows the Dryer Exhaust Fan #1, on which loose motor rotor bars were diagnosed, using this method. All vibration spectra were obtained using a Rockland Scientific Model 5815A Real Time Analyzer and a PCB Model 302 accelerometer. All data was analyzed using peak velocity spectra.

This unit was a large, belt-driven, overhung fan powered by a 350 HP AC induction motor. The client had obtained the number of rotor bars for each motor in the contract. Dryer Exhaust Fan No. 1 had 57 rotor bars and operated at a normal running speed of approximately 1795 RPM, resulting in a Rotor Bar Pass Frequency (RBPF) of approximately 102,315 CPM (1795 X 57 bars = 102,315).

II. CONCLUSIONS

Higher than normal 2X RBPF vibration levels were detected during the October, 1990 testing. Trending of the motor electrical problem was recommended at this time. As Figure 2 indicates, during the October, 1990 testing, one of the $2F_L$ sidebands was actually higher in amplitude than was 2X RBPF. No abnormalities were noted in the run-up data at this time.

The next scheduled RTA testing in February, 1991 was performed with the motor at normal running speed, only due to plant operating commitments. This data in Figure 2 indicates a significant increase in 2X RBPF and $2F_L$ sideband amplitude (from .017 in/sec to .163 in/sec at $2F_L$).

Testing performed in May, 1991 included the run-up testing and revealed that removal of the motor from service would be necessary to avoid catastrophic failure. Figure 2 is a comparison of 0 - 300,000 CPM spectrum from the October, February and May surveys and illustrates the continuing degradation of the rotor bar condition. Figure 3 shows the noticeable change from October to May in the 0 - 300,000 CPM peak-hold spectra made during the run-up of the machine. Figure 4 is a comparison of the run-up waterfall plots from Dryer Exhaust Fans No. 1 and No. 2, showing a motor in good condition (No. 2) and the No. 1 unit with loose rotor bars.

III. RESULTS & RECOMMENDATIONS

It was recommended that the motor be removed from service, disassembled and inspected for loose rotor bars. The motor was subsequently removed from service.

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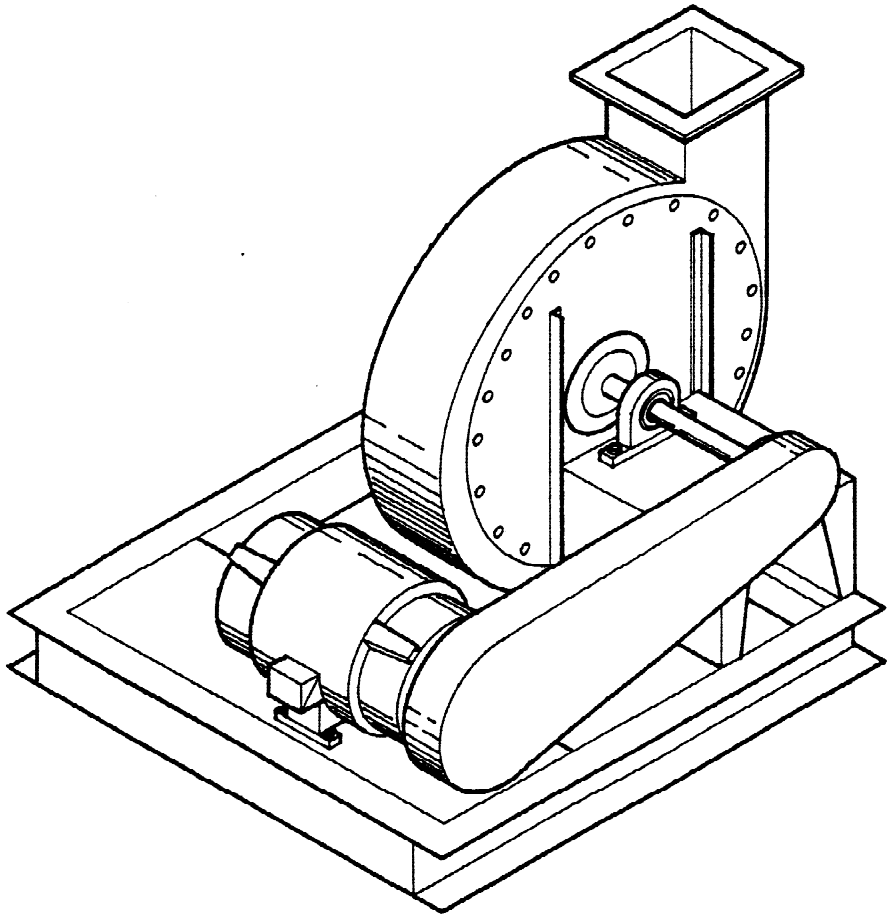
SIEMANS-ALLIS 350 H.P. MOTOR
57 ROTOR BARS
1795 RPM



DRYER EXHAUST FAN NO. 1

FIGURE 1

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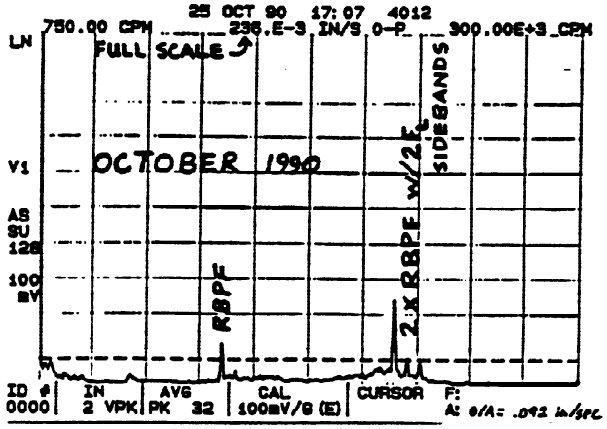
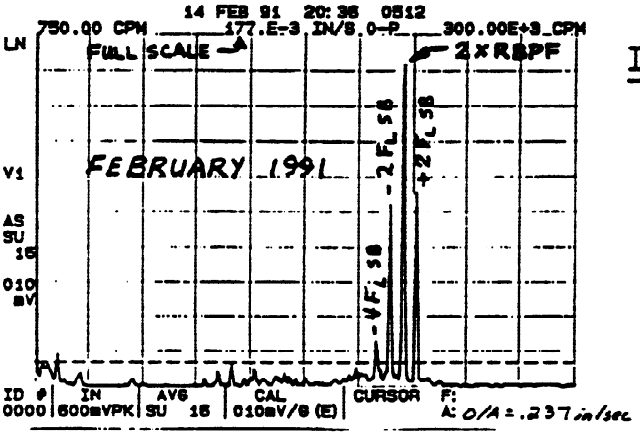
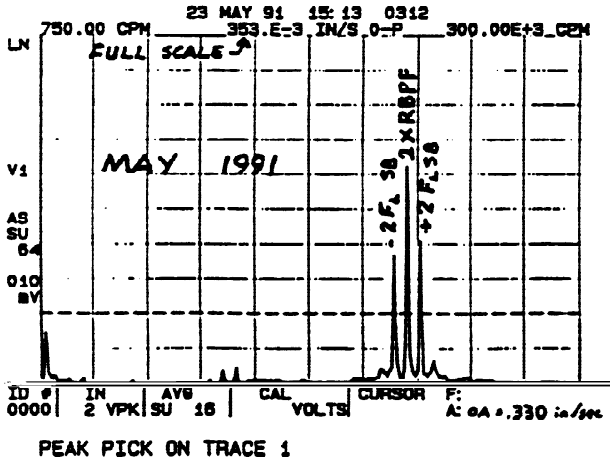


IN/S 0-P	CPM
26.9E-3 RBPF	102.00E+3
57.1E-3 -2F$_L$	197.24E+3
17.2E-3 2xRBPF	204.74E+3
16.2E-3 +2F$_L$	211.49E+3



IN/S 0-P	CPM
16.2E-3	12749.
22.6E-3 -4F$_L$	189.75E+3
92.9E-3 -2F$_L$	197.24E+3
163.E-3 2xRBPF	204.74E+3
98.6E-3 +2F$_L$	211.49E+3



IN/S 0-P	CPM
131.E-3 -2F$_L$	197.24E+3
221.E-3 2xRBPF	204.00E+3
145.E-3 +2F$_L$	211.49E+3

24-MAY-91 AMPLITUDE PROGRESSION OF 2X RBPF

12: 28: 48 MOTOR OUTBOARD BEARING, HORIZONTAL

FIGURE 2

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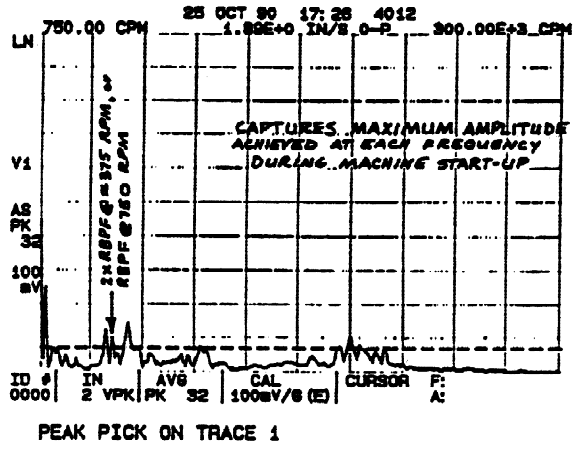
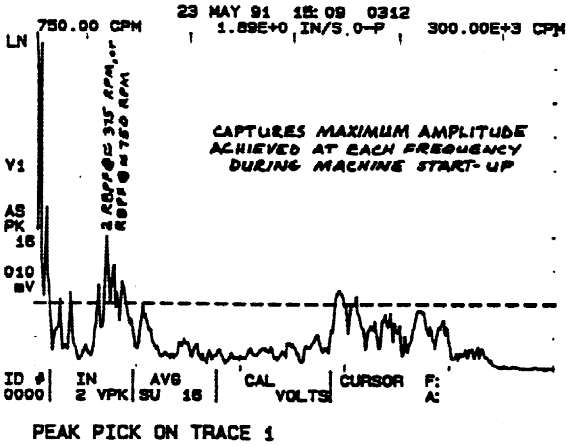


IN/S 0-P	CPM
475.E-3	3749.9
129.E-3	8250.0
233.E-3	39000.
181.E-3 1xRBPF@=375 RBPF@750 RPM	42750.
268.E-3	51748.
129.E-3	56248.
138.E-3	93000.
138.E-3	173.24E+3
190.E-3	180.00E+3
147.E-3	185.24E+3
138.E-3	199.50E+3

25-OCT-90 PEAK HOLD DURING MOTOR RUN-UP

17: 49: 21 MOTOR OUTBOARD BEARING, HORIZONTAL

D.EX.FAN#1



IN/S 0-P	CPM
1.84	3749.9
923.E-3	7500.0
406.E-3	15750.
440.E-3	21750.
483.E-3	38250.
759.E-3 1xRBPF@=375 RBPF@750 RPM	42750.
595.E-3	47250.
440.E-3	48748.
501.E-3	51748.
380.E-3	63750.
457.E-3	177.00E+3
380.E-3	180.74E+3
423.E-3	186.75E+3

30-MAY-91 PEAK HOLD DURING MOTOR RUN-UP

13: 11: 11 MOTOR OUTBOARD BEARING, HORIZONTAL

FIGURE 3

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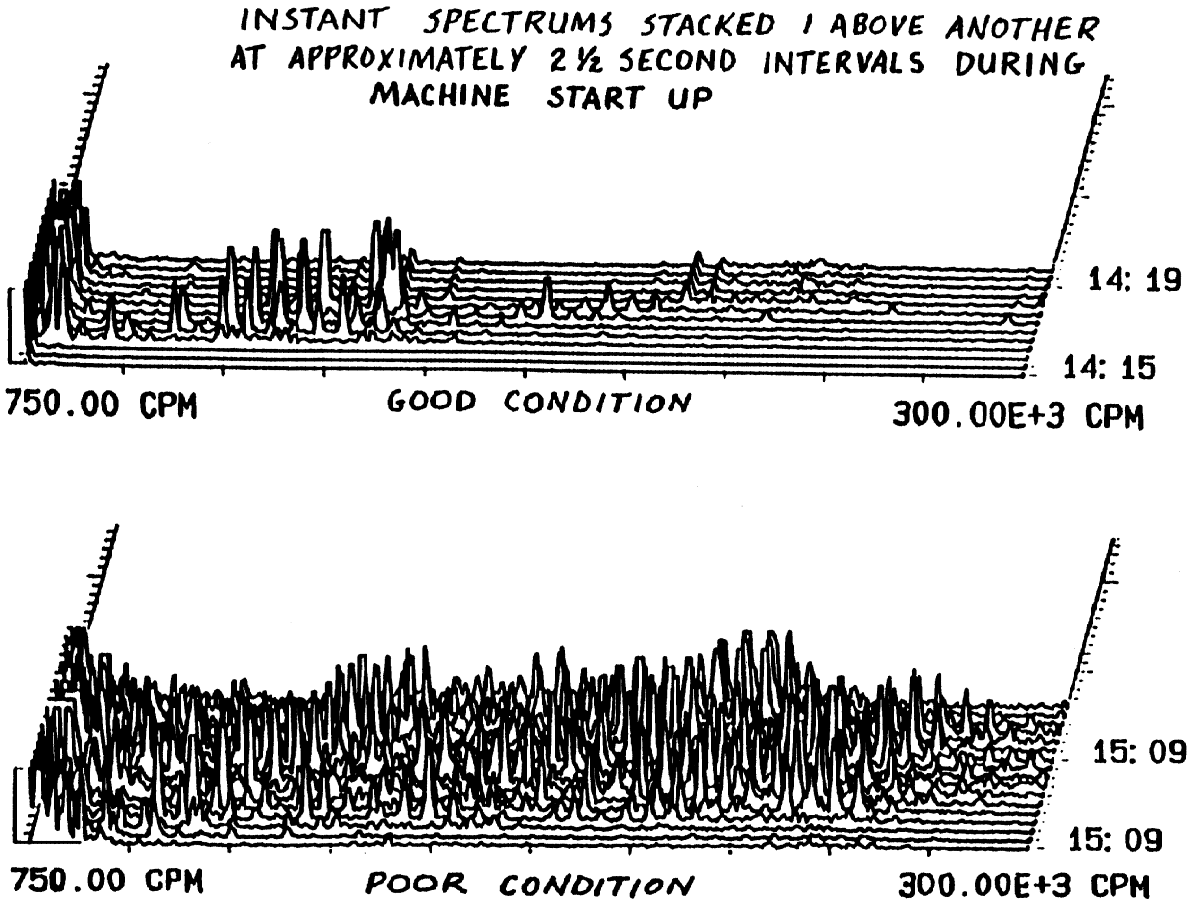

D.EX.FAN#2

D.EX.FAN#1

FIGURE 4

DETECTION OF LOOSE ROTOR BARS IN A PLOUGH SHEAR VARIABLE SPEED AC INDUCTION MOTOR

I. BACKGROUND

Technical Associates was given a contract to assist restructuring a PMP database to include correct vibration measurement locations, correct frequency ranges and spectral alarm bands. When the special motor electrical measurement positions of 1-0, Position 1 horizontal direction, 0 - 240,000 CPM and 2-0, Position 2 horizontal direction, 0 - 12,000 CPM were set up in the PMP route, new baseline data was collected and subsequently indicated an excessive vibration at 2X RBPF on the Plough shear motor. This machine was very critical to the process at this plant and was an inverter controlled, variable speed machine. It must be remembered that since this unit is inverter controlled, F_L and $2F_L$ are not equal to 3600 and 7200 CPM. F_L can be estimated by applying the actual speed/normal speed ratio to the normal F_L (3600 CPM). Figure 1 shows the machine's arrangement.

II. CONCLUSIONS/RECOMMENDATIONS

The motor was running at 900 RPM and is a 6 pole motor (normal 60 Hz speed = 1200 RPM). This indicated an F_L = 2700 CPM (900/1200 = .75; .75 X 3600 = 2700). Since the exact number of rotor bars was not known at the time, Figure 2 was carefully examined. The large, discrete peak at 97,200 CPM (Position 1, horizontal) was felt to be 2RBPF with multiple $2F_L$ (5400 CPM) sidebands. This would yield 54 rotor bars (97,200/2 = 48,600; 48,600/900 = 54) which was later confirmed by the motor manufacturer. Other motor position spectra were carefully evaluated and Position 2V (shown in Figure 3) was found to have a high vibration amplitude at RBPF (48,600 CPM) with F_L sidebands. This gave further support to a diagnosis of loose rotor bars. The amplitude at 2RBPF was .164 inches/second at Position 1-0 (horizontal) and the amplitude at RBPF was .105 inches/second at Position 2V. Both values are considerably above the alert level for RBPF of .060 inches/second.

Since no historical data on these vibration measurements were available, it was decided to collect data daily for the next week to determine if the motor condition was deteriorating. These daily measurements revealed that the motor condition was worsening and the amplitude of 2RBPF was steadily climbing. This is illustrated in the "waterfall" shown in Figure 4. It was determined at this time that removal and replacement of the motor was necessary. Note that the amplitude of 2RBPF increased from .164 inches/sec to .227 inches/sec the next day and then .314 inches/sec.

III. RESULTS OF CORRECTIVE ACTIONS

Figure 5 shows a waterfall plot of Position 2V illustrating the absence of 2RBPF following installation of the new motor. The initial report from the motor shop was that no problems could be found with the motor. The motor shop was then instructed to load the motor with a dynamometer and retest it. When this was done, the loose rotor bars became obvious.

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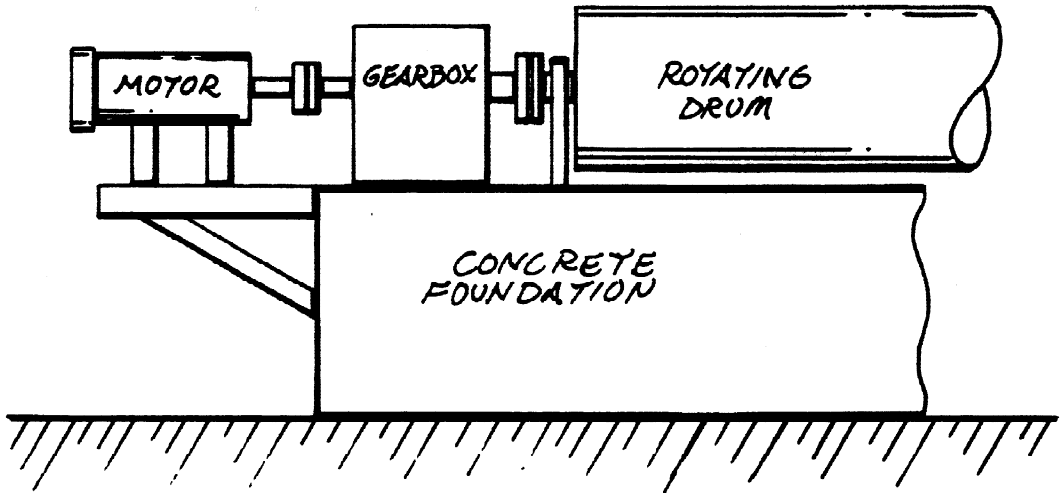

FL-569 PLOUGH

FIGURE 1

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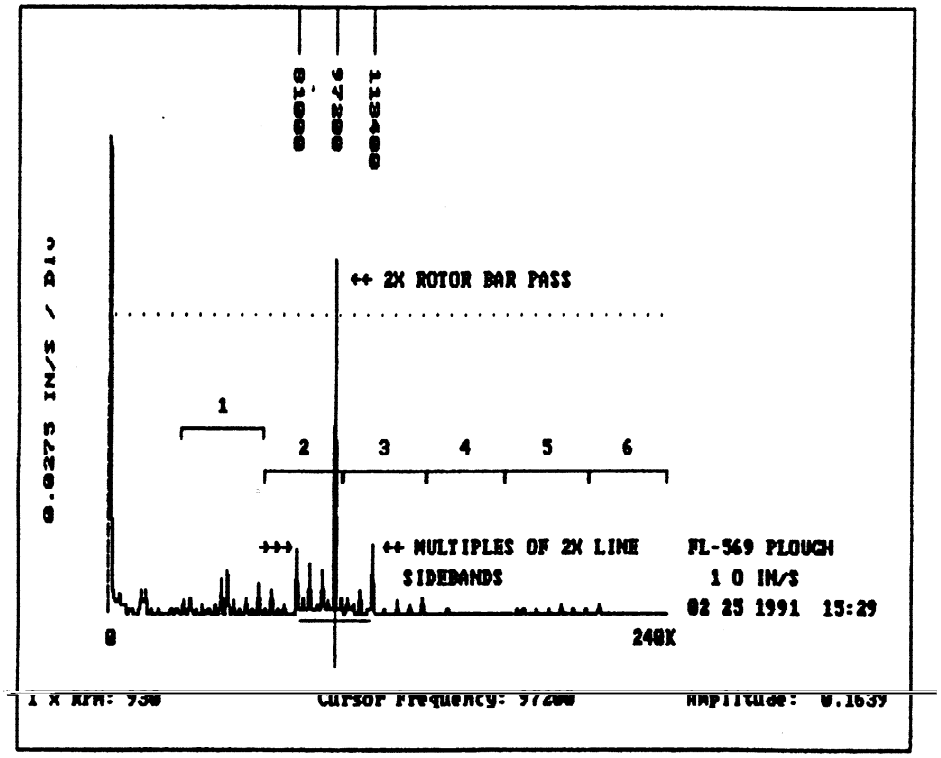

FIGURE 2

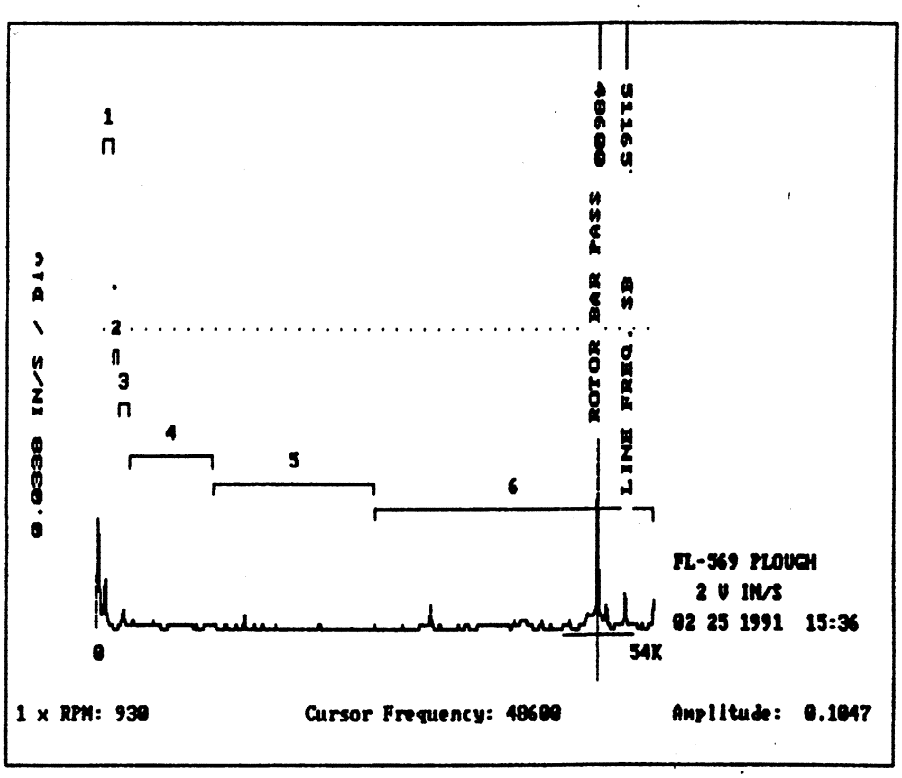

FIGURE 3

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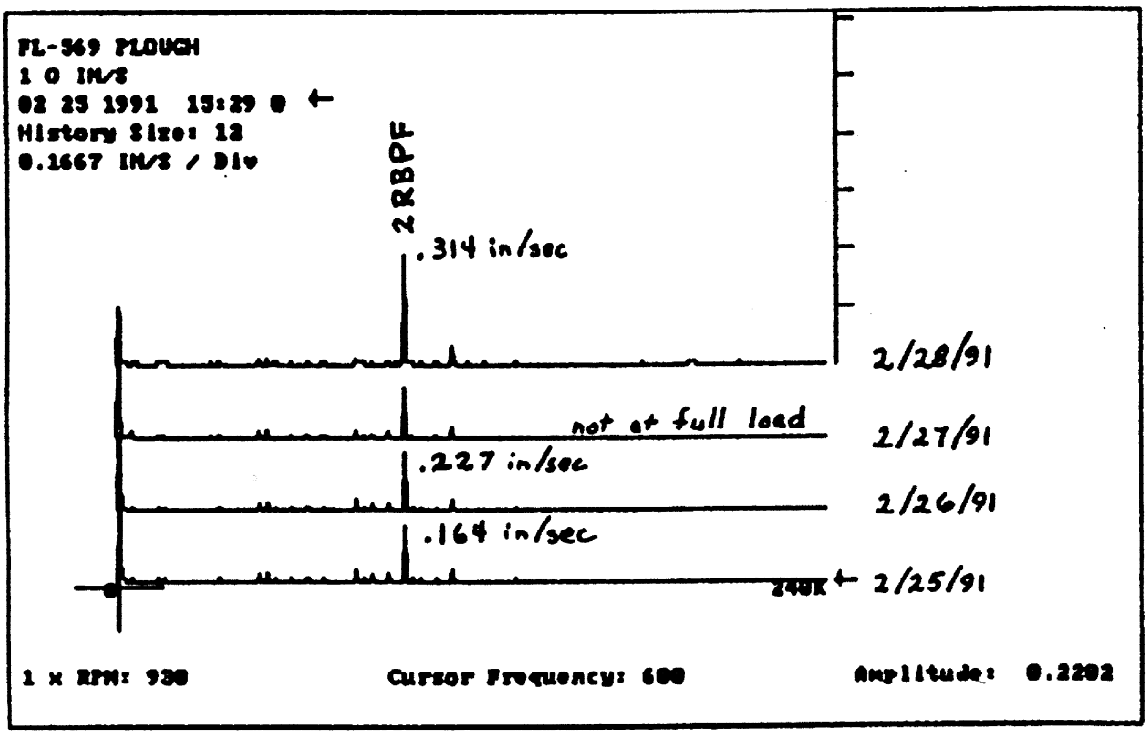

FIGURE 4

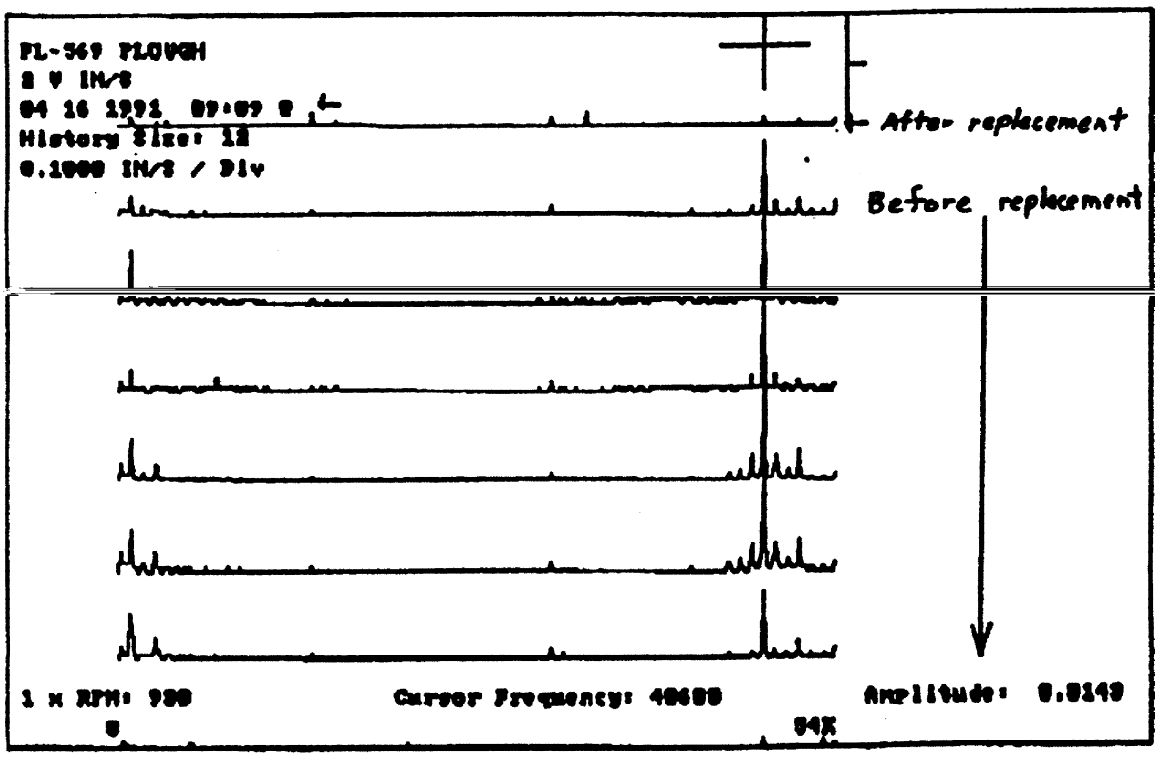

FIGURE 5

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DETECTION OF PHASING PROBLEM IN A VARIABLE SPEED CHILLER MOTOR

I. BACKGROUND

Technical Associates has a contract for conducting PMP surveys at a large manufacturing facility. Included in the equipment covered by this contract is a York centrifugal chiller powered by a variable speed electric motor. Figure 1 shows the machine drawing. During the normal PMP data collection the line frequency is adjusted to 60 Hz (F_L = 3600 CPM). The route data collected February 1989 indicated extreme vibration amplitudes at $2F_L$ (7200 CPM), as shown in Figure 2.

II. CONCLUSIONS/RECOMMENDATIONS

To verify that the problem was indeed at 2 F_L, the inverter frequency was adjusted to 53 Hz (F_L = 3180 CPM) and a 0 - 12,000 CPM, 3200 line FFT was collected. This data is shown in Figure 3 which shows a high amplitude at $2F_L$ (6390 CPM). Also shown in this spectrum are sidebands around $2F_L$ spaced at 1/3 F_L and a fundamental frequency of 1/3 F_L. A zoom on $2F_L$ as shown in Figure 4 more accurately defined the frequencies. $2F_L$ = 6371.2 CPM, and 1/3 F_L = 1095 CPM. The customer was informed (as well as the chiller manufacturer) that these vibration levels were unacceptably high and indicated that a problem affecting only one of the motor's phases likely existed.

III. RESULTS OF CORRECTIVE ACTIONS

The chiller manufacturer inspected the wiring connections at the motor junction box and reported that no problems were found. Further investigation was recommended and a loose crimp connection to one of the stator phase windings was located inside the motor housing with visible evidence of burning. The repair was made. Subsequent vibration readings indicated that the repair eliminated the vibration problem. The waterfall plot in Figure 5 illustrates this clearly.

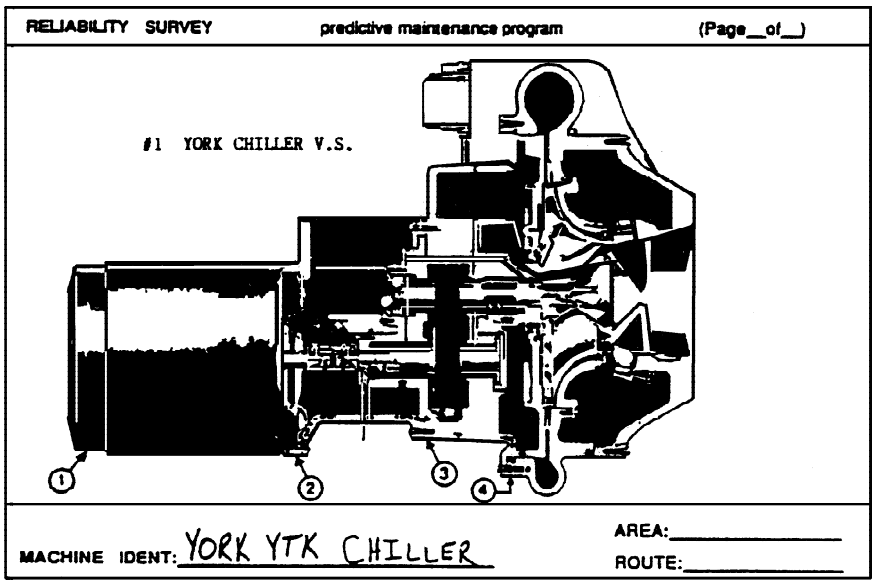

FIGURE 1.

M8 R-1092-1

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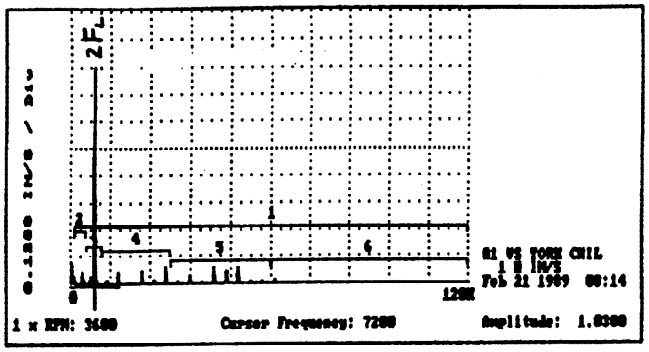


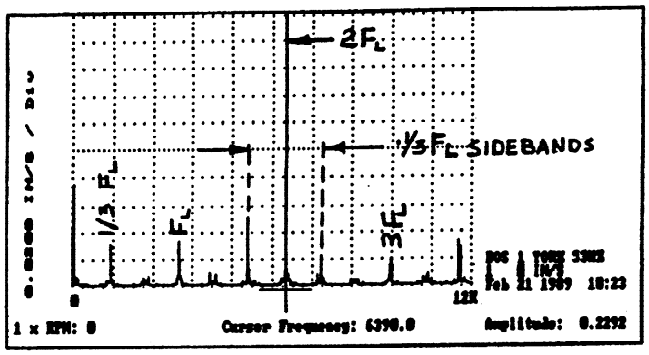
001% x 2.000 = 0.0200 1 x RPM: 3600.00

Machine	Pos	Dir	Units	Date
#1 VS YORK CHIL	1	H	IN/S	Feb 21 1989 08:1

Frequency	1 x RPM	Amplitude	
300	0.08	0.1001	
900	0.25	0.0299	
3600	1.00	0.0499	
6000	1.67	0.0299	
7200	2.00	1.0300	$2F_L$
11100	3.08	0.0200	
14400	4.00	0.0499	
21600	6.00	0.0600	
25200	7.00	0.0200	
28500	7.92	0.0699	
35700	9.92	0.0299	
42900	11.92	0.0699	
46800	13.00	0.0600	
50100	13.92	0.0699	
61200	17.00	0.0200	

T - Threshold adjust Q - Quit

FIGURE 2



004% x 0.200 = 0.0080 1 x RPM: ??????

Machine	Pos	Dir	Units	Date
BOS 1 YORK 53HZ	1	H	IN/S	Feb 21 1989 10

Frequency	1 x RPM	Amplitude	
30.0	-	0.0761	
1110.0	-	0.0319	$1/3\ F_L$
3180.0	-	0.0339	F_L
4140.0	-	0.0098	
4320.0	-	0.0085	
5280.0	-	0.0509	*-1/3 F_L SIDEBAND*
6390.0	-	0.2292	$2F_L$
6450.0	-	0.0089	
7350.0	-	0.0085	
7470.0	-	0.0221	*+1/3 F_L SIDEBAND*
9570.0	-	0.0196	$3F_L$
10680.0	-	0.0093	
11610.0	-	0.0336	
11670.0	-	0.0292	

FIGURE 3

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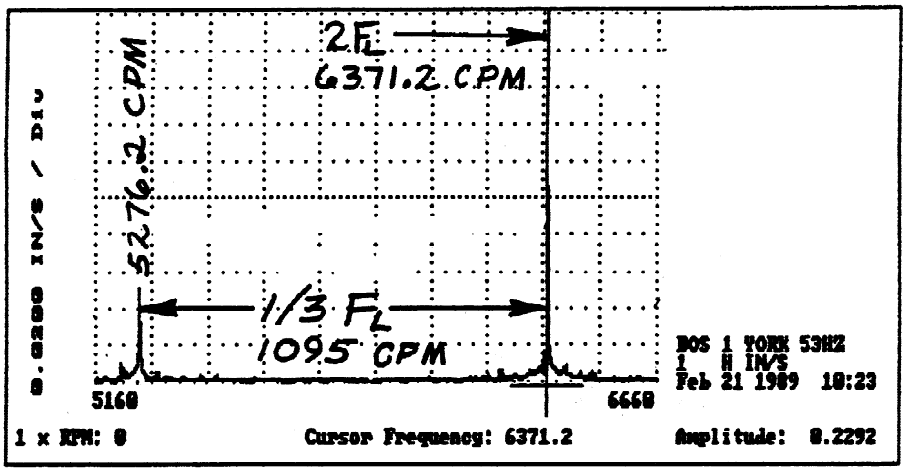

004% x 0.200 = 0.0080 1 x RPM: ??????

Machine	Pos	Dir	Units	Date
BOS 1 YORK 53HZ	1	H	IN/S	Feb 21 1989 10:

Frequency	1 x RPM	Amplitude	
5227.5	-	0.0098	
5276.2	-	0.0509	-1/3 FL SIDEBAND
6363.7	-	0.0166	
6371.2	-	0.2292	2FL
6423.7	-	0.0089	

FIGURE 4

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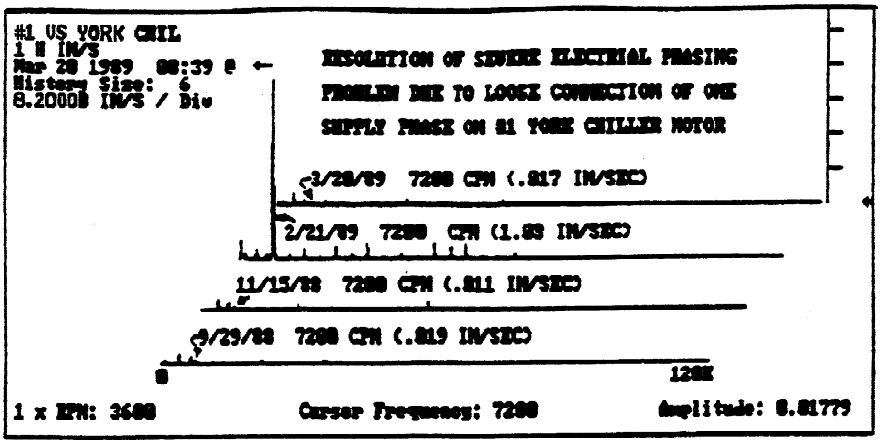
#1 VS YORK CHIL
1 H IN/S
Mar 20 1989 08:39 @
History Size: 6
8.2000B IN/S / Div
RESOLUTION OF SEVERE ELECTRICAL PHASING
PROBLEM DUE TO LOOSE CONNECTION OF ONE
SUPPLY PHASE ON #1 YORK CHILLER MOTOR
3/20/89 7200 CPM (.817 IN/SEC)
2/21/89 7200 CPM (1.83 IN/SEC)
11/15/88 7200 CPM (.811 IN/SEC)
9/20/88 7200 CPM (.819 IN/SEC)
0
120K
1 x RPM: 3600
Cursor Frequency: 7200
Amplitude: 0.81779

001% x 2.0000 = 0.02000 1 x RPM: 3600.00

Machine	Pos	Dir	Units	Date
#1 VS YORK CHIL	1	H	IN/S	Mar 20 1989 08:39

Frequency	1 x RPM	Amplitude
3600	1.00	0.06068
6000	1.67	0.02669
10800	3.00	0.02192
28500	7.92	0.02383
50100	13.92	0.02288

004% x 2.0000 = 0.08000 1 x RPM: 3600.00

Machine	Pos	Dir	Units	Date
#1 VS YORK CHIL	1	H	IN/S	Feb 21 1989 08:14

Frequency	1 x RPM	Amplitude
300	0.08	0.10008
7200	2.00	1.03002

001% x 2.0000 = 0.02000 1 x RPM: 3600.00

Machine	Pos	Dir	Units	Date
#1 VS YORK CHIL	1	H	IN/S	Nov 15 1988 13:09

Frequency	1 x RPM	Amplitude
3600	1.00	0.05274
6000	1.67	0.03272
21300	5.92	0.02097
49800	13.83	0.03781

000% x 2.0000 = 0.00000 1 x RPM: 3600.00

Machine	Pos	Dir	Units	Date
#1 VS YORK CHIL	1	H	IN/S	Sep 20 1988 09:35

Frequency	1 x RPM	Amplitude
300	0.08	0.00731
900	0.25	0.00699
3600	1.00	0.04988
6000	1.67	0.03590
7200	2.00	0.01875
10800	3.00	0.01334
21300	5.92	0.01843
35700	8.82	0.01174

FIGURE 5

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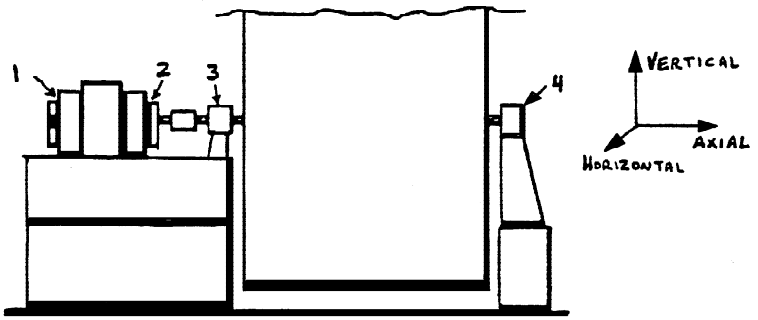
DETECTION OF MULTIPLE CRACKED ROTOR BARS WITHIN AN INDUCTION MOTOR USING BOTH VIBRATION AND MOTOR CURRENT ANALYSIS

When evaluating an especially critical motor, it is a good idea to combine both the motor current and vibration signature technologies using the results of both to diagnose potential problems and then to evaluate problem severity (not only in the motor, but also in the driven equipment). This particular example was such a case. This particular case involved a large induction draft (ID) fan at a power plant. In this case, the fan speed was periodically changed to blow the collected soot loose. During the speed increase, a loud whining sound was reported from within the motor and a 61,440 CPM (1024 Hz) response in excess of 4 g's was measured by personnel from the plant predictive maintenance team. In the past, the unit had triggered alarms shutting down the unit several times which was very costly to the utility. The driver was a 400 HP variable speed motor with nameplate data of 885 RPM, 467 Amps, 460 volts, 60 Hz. The motor was reported to have 70 rotor bars and was outfitted with SKF 6222 ball bearings. Figure 1 shows where measurements were performed on both the motor (Positions 1 & 2) and fan (Positions 3 & 4). As the drawing shows, measurements were made in horizontal, vertical and axial directions. While at the unit, both a complete vibration and phase analysis was performed and measurements were captured on a 4-channel tape recorder. Both vibration and current signature analyses were performed using a narrowband, real-time analyzer. In the case of vibration measurements, both velocity and acceleration measurements were captured and a number of tests were performed including natural frequency, impulse, runup and coastdown-type evaluations.

FIGURE 1. MEASUREMENT LOCATIONS FOR VIBRATION TEST



MACHINE IDENT: ID FAN #2 AREA: _____ ROUTE: _____

DRIVER	~~INTERMEDIATE~~	DRIVEN
RPM min: 885 (VARIABLE)	RPM in : _____	RPM min : _____
max : _____	out : _____	max : _____
MFGR : SIEMANS-ALLIS	RATIO: _____	MFGR. : _____
MODEL 124 FRAME: 5111AS TYPE: RA2CH	MODEL : _____	MODEL : _____
SER NO. : 7-5113-39068-1-1	SER NO. : _____	SER NO. : _____
V 460 P ____ A 467	RATING : _____	RATING : _____
~~SHEAVE~~ : _____ DIA.	BELT # : _____	#BLADES : _____
BRG#: SKF 6222 (I) Pos 2	BRG#: _____ (I)	BRG#: SLEEVE (I) Pos 3
BRG#: SKF 6222 (O) Pos 1	BRG# _____ (O)	BRG#: SLEEVE (O) Pos 4
H.P.: 400	GEAR T: _____ CT.	~~GEAR T~~: _____ CT.
OTHER: 70 ROTOR BARS, CAST	OTHER: _____	OTHER: _____

R-1092-1

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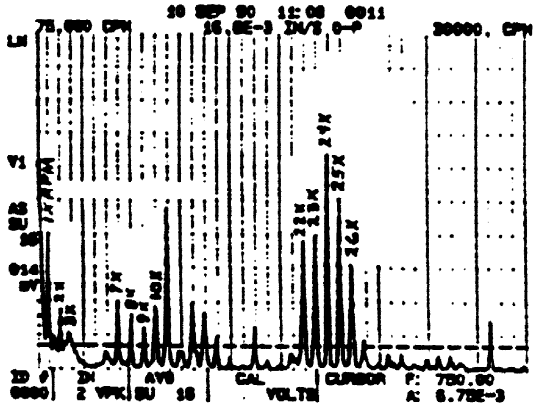
Table I in Section 1.0 summarizes each of the major signature patterns involved with electrical problem detection using vibration analysis. Comparing these signature patterns with those of current signature analysis, the two technologies have some similarities, but also some differences. For example, in the case of rotor bar problems, vibration analysis looks for pole pass frequencies particularly surrounding 1X RPM, while motor current analysis looks for the same pole pass frequencies, but centered around the line frequency component. Likewise, relative heights of the sidebands to the primary frequencies are of importance (this aspect of this signature is of greater importance in the case of current analysis).

Differing somewhat is where a variable air gap problem is found. In the case of vibration analysis, one looks primarily in the region of 2X line frequency, particularly looking for high amplitudes at this peak and whether or not it is surrounded with pole pass frequencies. However, vibration measurements do look much higher in frequency in the region of rotor slot frequency for what is known as the rotor bar pass frequency (# rotor bars X RPM). This is still within the general vicinity of f_{AG}, and in both cases, one is primarily looking for 2X line frequency sidebands spaced around the primary frequency (f_{AG} or RBPF).

Getting back to the example, neither vibration or current signature analysis found problems with air gap between the rotor and stator. However, both technologies found rotor bar problems.

Figure 2 shows a wide band spectrum taken from 0 to 30,000 CPM (500 Hz) and showed a myriad of motor running speed harmonics throughout the spectrum. A series of "zoom spectra" were then taken around the lower running speed harmonics between 1X and 3X RPM; medium speed harmonics between 7X and 12X RPM; and the higher harmonics between 22X and 26X RPM. Figure 5 shows one of these zoom spectra which was concentrated between 22X and 25X RPM. Note that even at these high running speed harmonics, each of them were surrounded by series of pole pass frequency sidebands spaced at about 60 CPM (actual = 60.6 CPM = 1.009 Hz). Therefore, with the significant number of pole pass frequencies (P) sidebands throughout the spectra, definite rotor bar problems were indicated.



IN/S RMS	CPM
6.75E-3	750.00
1.61E-3	1050.0
2.99E-3	1500.0
1.30E-3	1800.0
1.84E-3	2024.9
1.46E-3	2250.0
3.38E-3	5175.0
2.45E-3	6000.0
2.07E-3	6749.7
3.15E-3	7424.7
8.05E-3	8174.7
3.30E-3	9675.0
2.84E-3	10425.
1.61E-3	11174.
2.15E-3	13424.
6.44E-3 22X RPM	16425.
6.67E-3 23X RPM	17175.
10.8E-3 24X RPM	17925.
8.59E-3 25X RPM	18675.
5.29E-3 26X RPM	19350.

FIGURE 2

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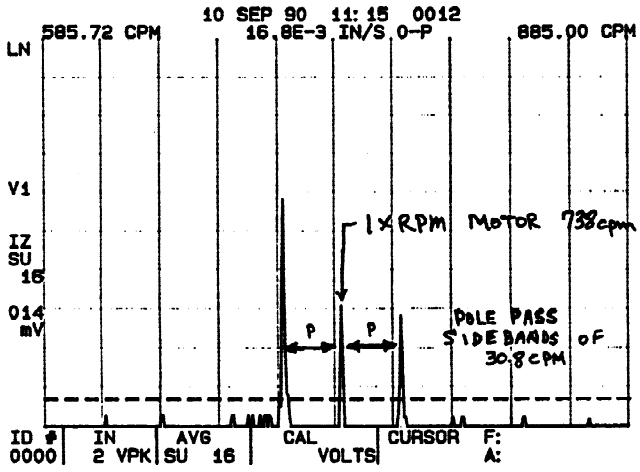


PEAK PICK ON TRACE 1

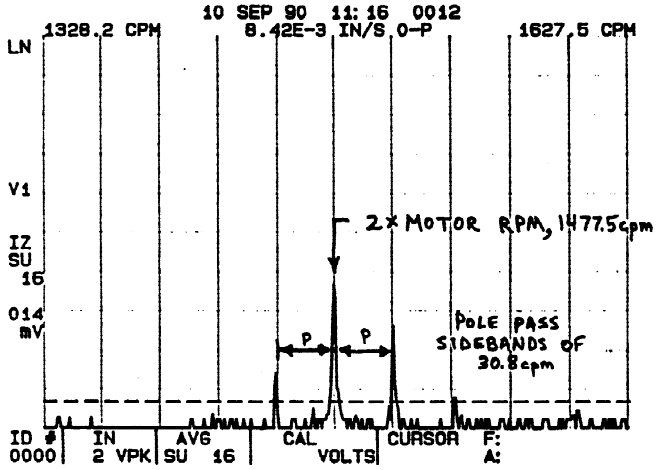
IN/S RMS		CPM
1.30E-3		705.75
9.82E-3		708.00
5.22E-3	1X MOTOR RPM	738.75
4.76E-3		769.50

12-SEP-90

15: 14: 17

POSITION 1-H
(MOTOR OUTBOARD
HORIZONTAL)
ID FAN #2

FIGURE 3



PEAK PICK ON TRACE 1

IN/S RMS		CPM
1.19E-3		1446.7
3.11E-3	2X MOTOR RPM	1477.5
2.23E-3		1508.3
654.E-6		1539.0

12-SEP-90

15: 15: 55

POSITION 1-H
(MOTOR OUTBOARD
HORIZONTAL)
ID FAN #2

FIGURE 4

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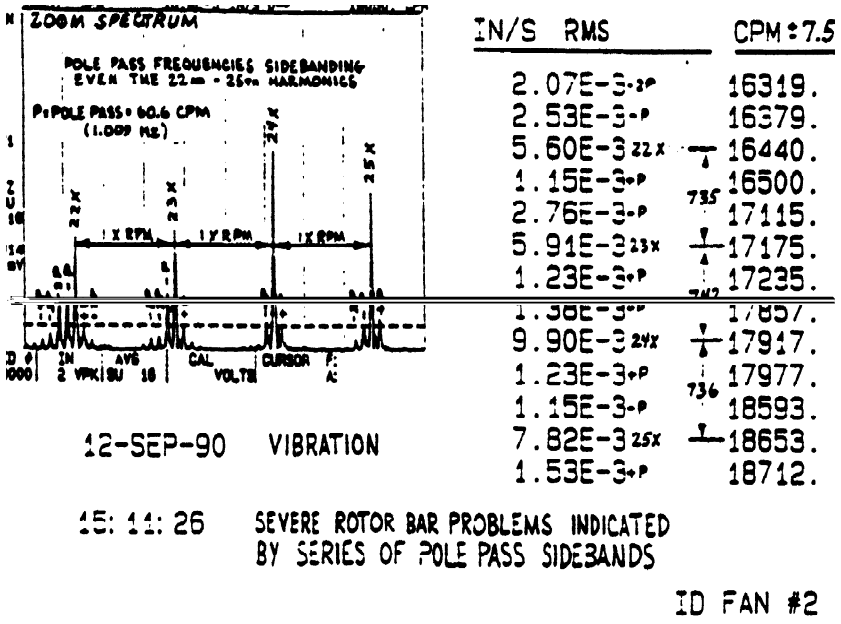

FIGURE 5

Next, some current spectra were taken from each of 3-phase leads for the #2 motor serving this #2 ID Fan. One of these spectra is shown in Figure 6. Note the line frequency (f_L) at 51.023 Hz. Also note the log amplitude scale which showed the line frequency to equal 1.89 "engineering units" compared to the pole pass sideband amplitude of .051 "engineering units" (due to the rush of time, no attempts were made to calibrate the current to a 0 dB rating). Figure 6 shows how the dB calculation was made. Note the equation on the drawing alone with the values entered. The resultant showed that the ratio of f_L/f_{SB} was only 31.4 dB (that is, f_L was only 31.4 dB higher than f_{SB}). Referring to Table I, this put this motor in category 6 which indicated "multiple cracked or broken rotor bars and end rings".

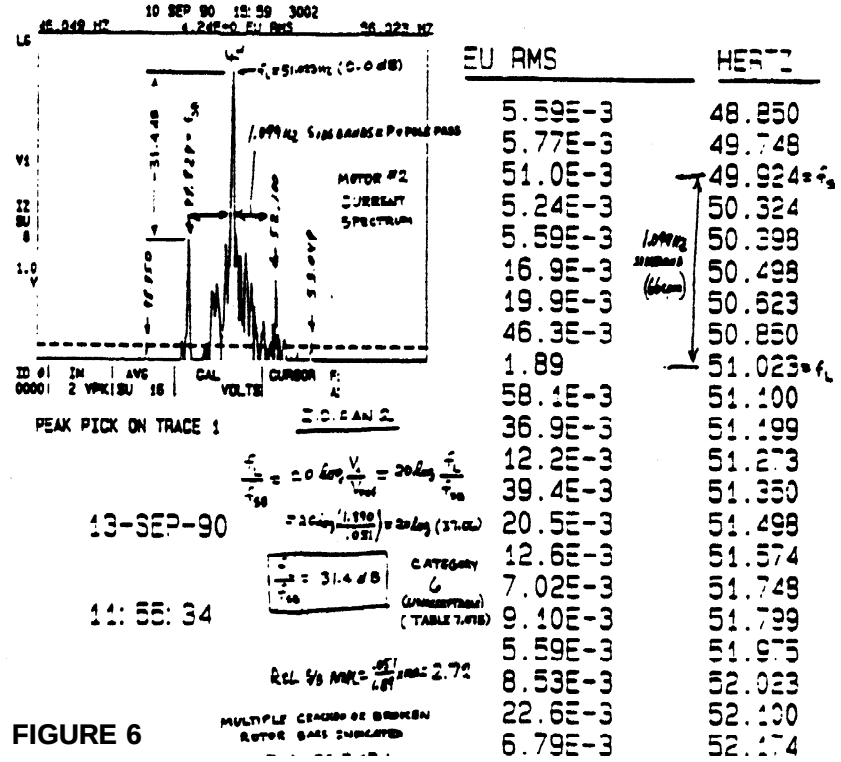

FIGURE 6

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While the analysis was being performed, the area superintendent asked if a similar analysis could quickly be done on motor #4 on which he had been hearing some "strange noises". Therefore, a similar current spectrum analysis was performed which is shown in Figure 7. Although this motor showed a f_L/f_{SB} ratio of 33.7 dB (slightly better than that of motor #2), this motor #4 was felt to be in even worse condition than motor #2 due to the second family of pole pass sidebands to the left and right of line frequency. Before leaving, similar tests were run on two other motors in the plant with much more favorable results.

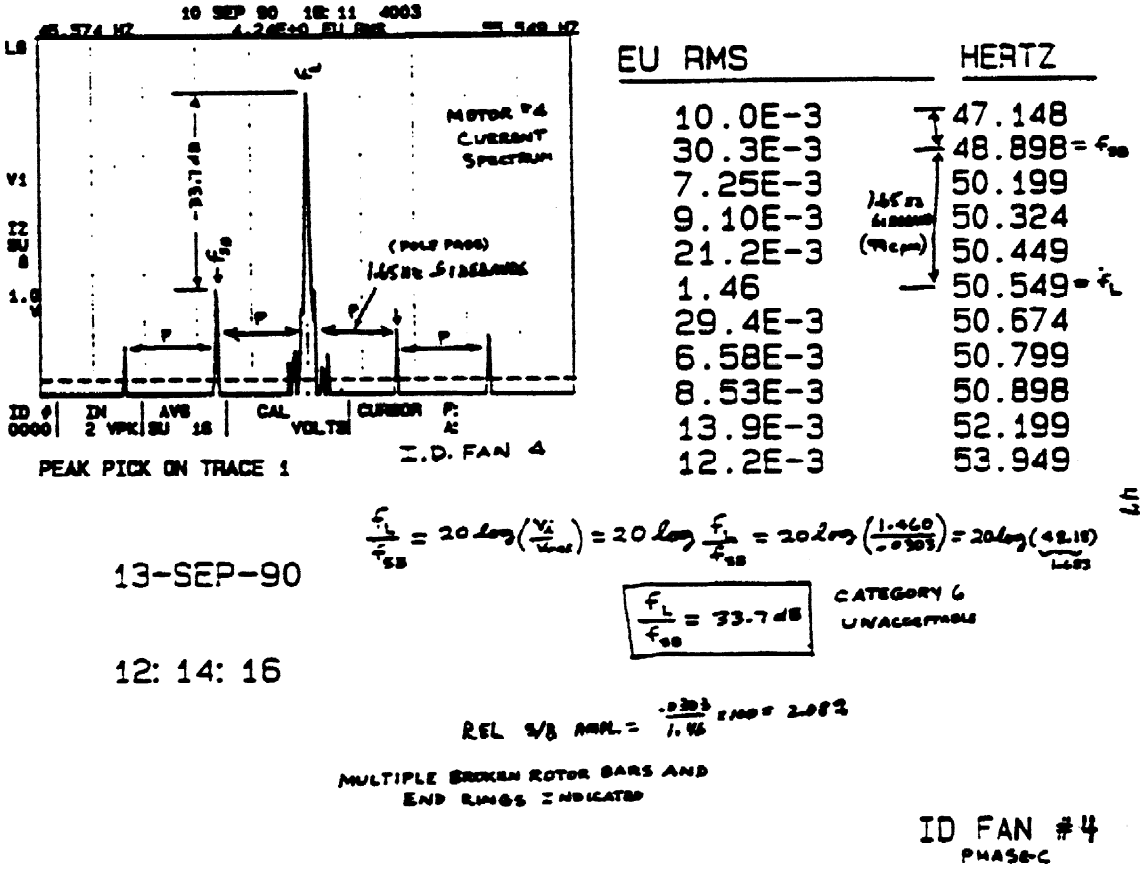

FIGURE 7

Technical Associates recommended immediate inspection and overhaul of both motors. Checking back with the client later, it was learned that after opening up the motors, the repair shop recommended a "nearby cemetery" for each motor. Subsequently, each motor was "laid to rest". In both cases, a series of rotor bars were broken and/or cracked and considerable damage was done to the end rings. Motor #4 also suffered considerable damage to the rotor windings and had some indications of contact with the stator.

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At this time, the author will offer his **opinion**, but reserves the right to refine these with further experience. This suggested motor current analysis severity chart for assessing rotor condition is given in Table I:

TABLE I

MOTOR CURRENT ANALYSIS SEVERITY CHART FOR ASSESSING ROTOR CONDITION WHEN COMPARING LINE CURRENT & LEFT POLE PASS FREQUENCY SIDEBAND AMPLITUDES (1)

CAT. NO.	f_L/f_{SB} dB	f_L/f_{SB} RATIO	f_{SB}/f_L RATIO %	ROTOR CONDITION ASSESSMENT	RECOMMENDED CORRECTIVE ACTION
1	>60 dB	>1000	<0.10%	Excellent	None
2	54-60	501-1000	.10-.20%	Good	None
3	48-54	251-501	.20-.40%	Moderate	Continue Surveys; Trend Only
4	42-48	126-251	.40-.79%	Rotor Bar Crack May Be Developing or Problems with High Resistance Joint(s)	Reduce Survey Intervals; Trend Closely
5	36-42	63-126	.79%-1.58%	Two Rotor Bars Likely Cracked or Broken & Problems with High Resistance Joints Likely	Perform Vibration Tests to Confirm Problem Source & Severity
6	30-36	32-63	1.58-3.16%	Multiple Cracked or Broken Rotor Bars & End Rings Indicated; Also Slip Ring & Joint Problems	Overhaul ASAP
7	<30 dB	<32	>3.16%	Multiple Broken Rotor Bars & End Rings Very Likely; Severe Problems Throughout	Overhaul or Replace ASAP

NOTE:

(1) Minimum Induction Motor Load = 80%

(2) f_L = Amplitude of Line Frequency (dB or Amps)

(3) f_{SB} = Amplitude of 1st Pole Pass Sideband to Left of Line Freq.(dB or Amps)

(4) Pole Pass Frequency = # Poles X Slip Frequency (Hz or CPM)

(5) Slip Frequency = $\frac{\text{2X Line Freq}}{\text{\# Poles}}$ __ MOTOR RPM

(6) "CAT NO." = Category Number (or Severity Ranking)

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DETECTION OF ELECTRICAL PROBLEMS WITHIN A SYNCHRONOUS MOTOR SERVING A RECIPROCATING AIR COMPRESSOR DUE TO LOOSE COILS

I. BACKGROUND

The drive motor for Compressor No. 3 was a 400 HP G.E. Synchronous motor, 514 RPM, Serial No. LJ8374536, model No. 5TS850014A3. The compressor had two 90° opposed pistons and was an Ingersoll-Rand XLE 24 for 110 psi service. Discussions with General Electric indicated that there were 14 poles in the motor and 6 coils per pole for a total of 84 coils. Measurement locations are shown in Figure 1.

A number of motor failures had occurred on No. 3 Compressor involving the stator on one occasion and the rotor on another. A vibration investigation was performed on January 26, 1990 in an attempt to determine the health of the motor. Measurements were also obtained on Compressors 4, 5 and 6 for comparison to No. 3. These comparisons were helpful in diagnosing the problem. If problems were found, recommendations to resolve such problems were requested.

II. CONCLUSIONS

Figure 2 is a 60,000 CPM spectrum taken on motor Position 1V and showed many harmonics of running speed which was an indicator of possible bearing looseness or excessive clearances. A check of bearing clearances and tightness of bearings was recommended.

The vibration signatures for Compressor No. 3 also indicated motor electrical problems. Figure 2 also showed a noticeable peak at 2X RPM which responds to secondary reciprocating forces in these XLE compressor design types (1X RPM responds to the primary forces). Probably the most important item in the plot was the peak at 43,200 CPM which was 84 times running speed of the motor (measurement locations are shown on Figure 1). The dominant high frequency spike on Figure 2 was found to occur only on Compressor No. 3. Discussions with G.E. indicated that the motor on Compressor No. 3 had 14 poles with 6 coils per pole which gave a total of 84 coils. Since the high frequency spike occurred at 84 times motor speed, this is the coil pass frequency. Therefore, it was likely that the motor had loose coils. Figure 3 is a high resolution zoom on the 84X frequency which showed motor running speed sidebands. The motor speed sidebands were an indicator that the 84X was associated with the 510 CPM rotor, and not some other source.

Load tests were conducted to provide further evidence of electrical problems (if the vibration noticeably drops with a decrease in load, electrical problems such as loose coils within a synchronous motor are indicated). Load tests indicated that the amplitude of the 84X frequency did drop significantly as the load decreased. This is seen in Figures 4, 5, 6 and 7.

Discussions with G.E., the motor manufacturer, indicated that loose stator coils are more common on reciprocating compressors than in other motor applications and, once the synchronous motor coils become loose, the problem can propagate quickly resulting in early failure. Since maximum vibration occurred at 84X running speed (Coil Passing Frequency) the source was likely loose coils.

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III. RECOMMENDATIONS

1. It was recommended that the compressor bearings be checked for excessive wear and replaced if clearances were outside of specified limits.

2. It was also recommended that the motor be checked for loose stator coils. If loose stator coils were found, it was recommended that OEM specifications be used in which special materials and procedures could be applied to ensure proper coil wedging since coil looseness is a recognized potential problem in this type of application. G.E. felt if the coils were found loose, they could attempt to rectify this by pouring an epoxy-type liquid into each pole which would hopefully fill as many voids as possible which had been created over the years of operation.

IV. RESULTS OF CORRECTIVE ACTIONS

The motor was sent to the G.E. service shop in Charlotte, NC where the coils were indeed found to be loose. Due to the motor's configuration, the G.E. service people were not able to directly access the coils for a foolproof repair. However, they were able to pour an epoxy material into the gaps and fill the voids in each pole as best they could (each pole was outfitted with a bundle of 6 coils).

Upon reinstallation of the repaired rotor, vibration at the coil pass frequency (84X RPM) reduced from .134 in/sec to only .059 in/sec at 84X RPM (CPF) as seen by comparing Figures 8 and 9. Once again, the load tests were repeated to evaluate the sensitivity of Coil Pass Frequency vibration to load. Now, instead of seeing a drop from .134 in/sec loaded to .080 in/sec unloaded, CPF dropped only from .059 in/sec loaded to .046 in/sec unloaded. This signaled a noticeable improvement in electrical operating condition for this synchronous motor.

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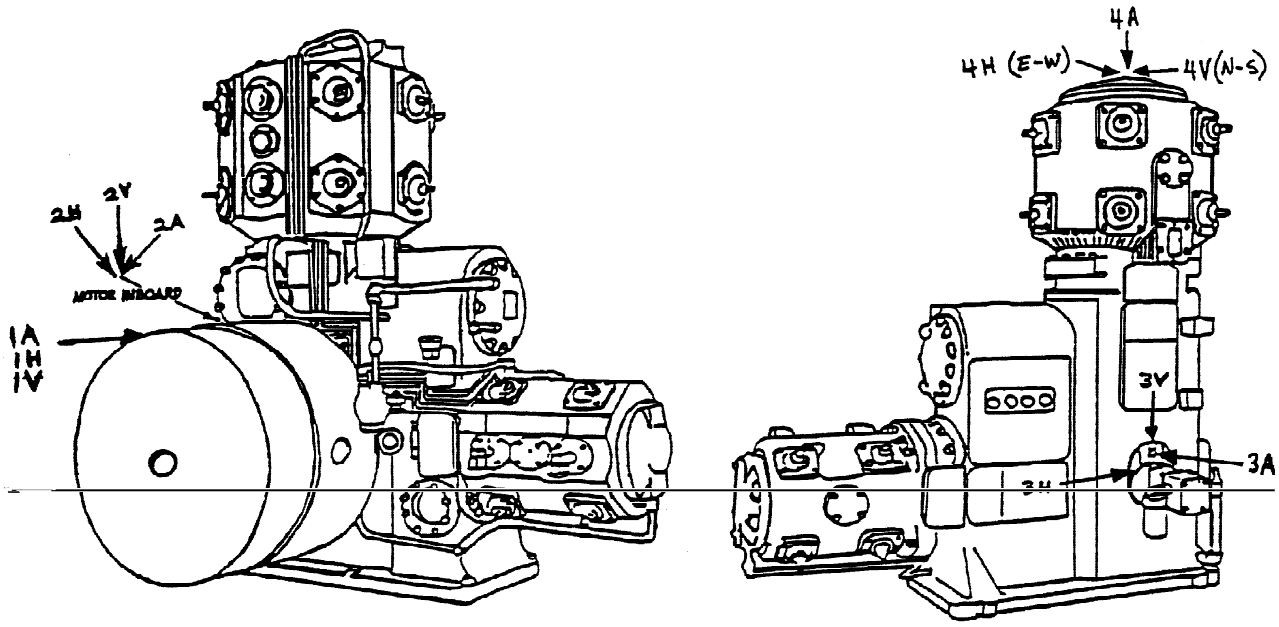

GE Synchronous 400 HP Motor, 514 RPM
Ingersoll-Rand XLE Compressor #3

FIGURE 1

MEASUREMENT LOCATIONS ON XLE #3 MOTOR AND COMPRESSOR

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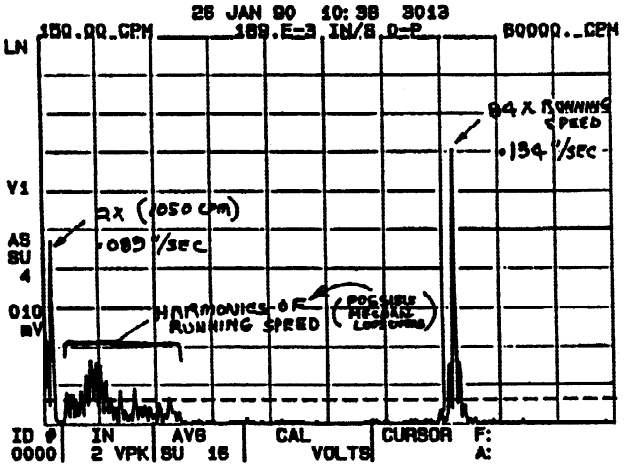
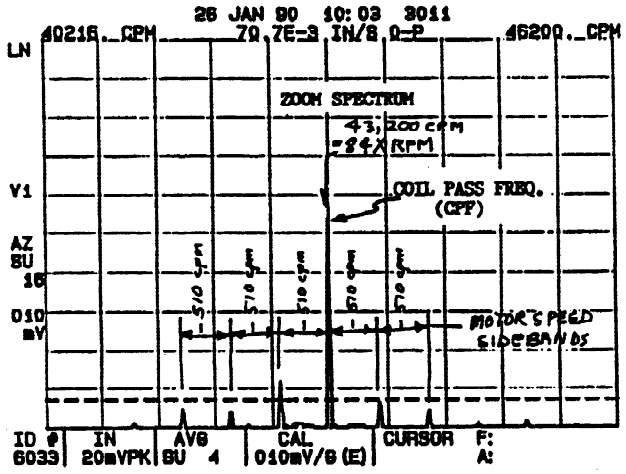


IN/S 0-P		CPM
40.6E-3		450.00
88.9E-3	2xRPM -	1050.0
14.7E-3		2550.0
13.8E-3		3000.0
15.5E-3		4049.9
23.3E-3		4650.0
31.1E-3		5100.0
29.3E-3		5700.0
28.5E-3		6149.7
20.7E-3		6749.7
12.1E-3	2xFINE -	7200.0
15.5E-3		8250.0
16.4E-3		9750.0
12.1E-3		13349.
28.5E-3		42600.
134.E-3	84xRPM -	43198.
30.2E-3		43650.

COMPRESSOR No. 3
FULL LOAD - 450 AMPS
POSITION IV

XLE #3

FIGURE 2



IN/S 0-P		CPM
8.07E-3		42690.
38.1E-3	84xRPM -	43200.

510 CPM SIDEBAND

COMPRESSOR NO.3
FULL LOAD - 450 AMPS
POSITION 1A

FIGURE 3

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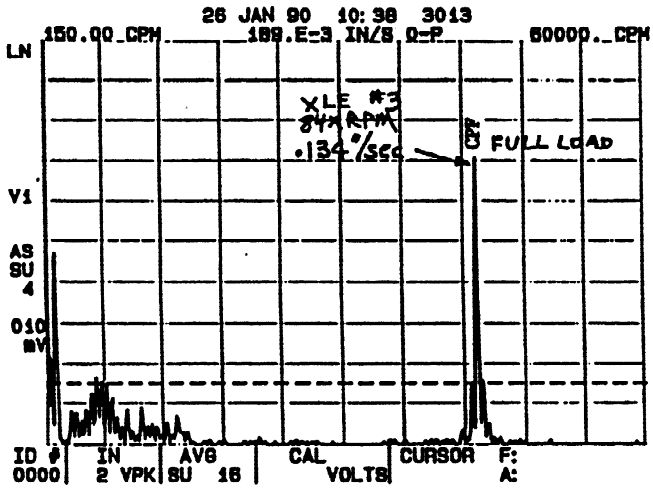
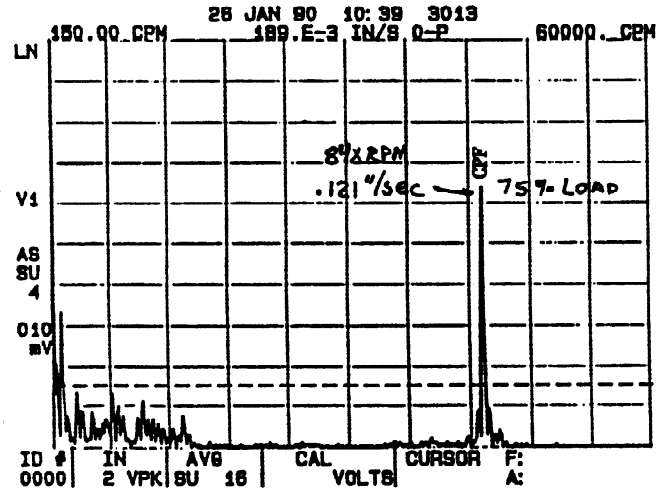


IN/S 0-P	CPM
40.6E-3	450.00
88.9E-3	1050.0
31.1E-3	5100.0
29.3E-3	5700.0
28.5E-3	6149.7
28.5E-3	42600.
134.E-3	43198.
30.2E-3	43650.

COMP. NO. 3
FULL LOAD, POSITION IV
450 AMPS
LOAD TEST

XLE #3

FIGURE 4
FULL LOAD



IN/S 0-P		CPM
38.8E-3		450.00
63.0E-3		1050.0
121.E-3	84%RPM-	43198.

OA = .158
COMP. NO. 3
3/4 LOAD, POSITION IV
310 AMPS

FIGURE 5
75% LOAD

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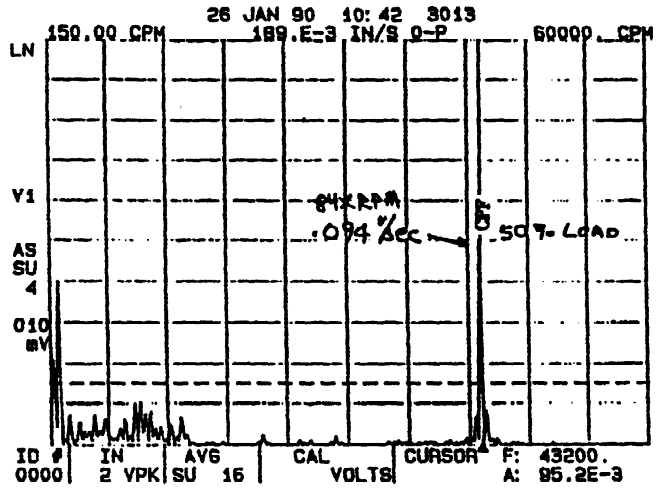
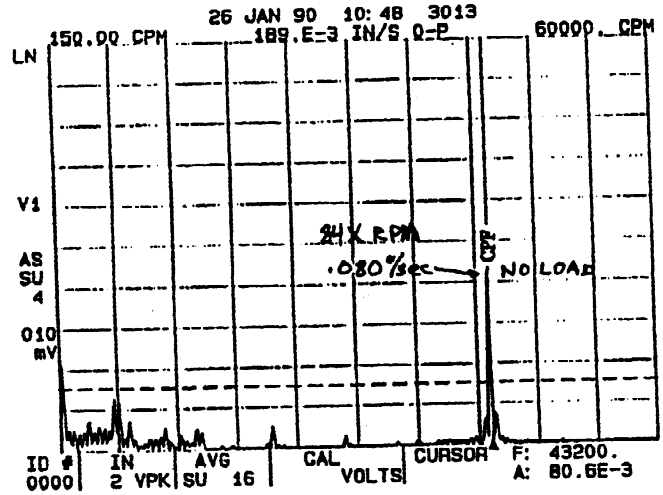


IN/S 0-P	CPM
39.7E-3	450.00
75.1E-3	1050.0
94.9E-3	84xRPM-43198.

OA = .137
COMP. NO. 3
1/2 LOAD, POSITION IV
200 AMPS

XLE #3

FIGURE 6
50% LOAD



IN/S 0-P	CPM
39.7E-3	450.00
80.3E-3	84xRPM-43198.

OA = .100
COMP NO. 3
NO LOAD, POSITION IV
130 AMPS

FIGURE 7
NO LOAD

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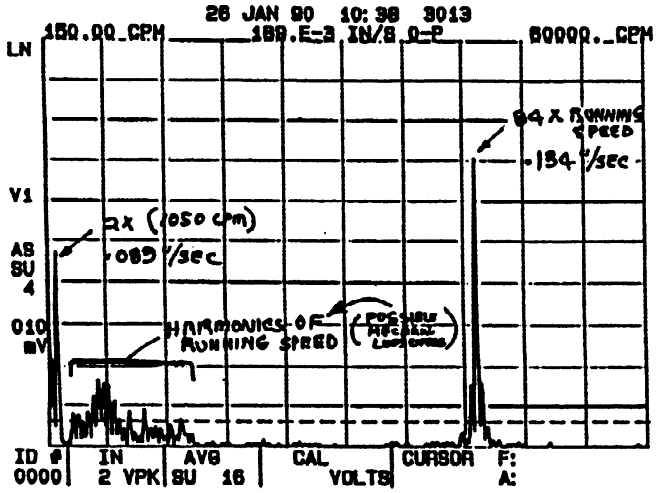
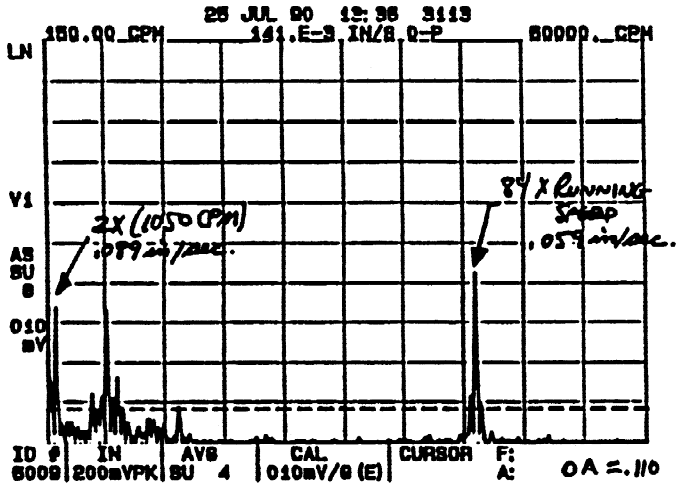


IN/S 0-P		CPM
40.6E-3		450.00
88.9E-3	2xRPM -	1050.0
14.7E-3		2550.0
13.8E-3		3000.0
15.5E-3		4049.9
23.3E-3		4650.0
31.1E-3		5100.0
29.3E-3		5700.0
28.5E-3		6149.7
20.7E-3		6749.7
12.1E-3	2x Fine -	7200.0
15.5E-3		8250.0
16.4E-3		9750.0
12.1E-3		13349.
28.5E-3		42600.
134.E-3	84xRPM-	43198.
30.2E-3		43650.

COMPRESSOR No. 3
FULL LOAD - 450 AMPS
POSITION IV
JAN. 26, 1990

XLE #3

FIGURE 8
LOOSE COILS IN SYNCHRONOUS MOTOR (BEFORE SPECTRUM)
(14 POLES X 6 COILS/POLE = 84 X RPM)



IN/S 0-P	CPM
21.9E-3	450.00
47.6E-3	1050.0
16.7E-3	4650.0
11.6E-3	5100.0
16.1E-3	5700.0
47.0E-3	6149.7
15.5E-3	6749.7
22.5E-3	7200.0
12.2E-3	7650.0
12.2E-3	13349.
16.1E-3	42748.
59.9E-3	43198.
13.5E-3	43650.

FULL LOAD - 420 AMPS
COMPRESSOR No. 3
POS 1 VERTICAL
JULY. 25, 1990

XLE #3

FIGURE 9
RESOLUTION OF LOOSE COIL PROBLEM (AFTER REPAIR)

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SIGNIFICANT IMPROVEMENT IN MOTOR AND BLOWER VIBRATION DUE TO BELT REPLACEMENT

I. BACKGROUND

During a Predictive Maintenance Program vibration survey by Technical Associates on March 22, 1989, a belt-driven air handler, 304 AIRHAND 004, was detected as having an overall vibration of 1.34 in/sec on Position 2H. Position 2H is the inboard motor bearing horizontal measurement as seen in the machine sketch of Figure 1. Note the Figure 1 spectrum for Position 2H on March 22, 1989 shows the dominant vibration was .930 in/sec at 450 CPM. Motor speed was a nominal 1800 RPM and fan speed a nominal 1050 RPM as verified by a hand-held tachometer measurement. It was determined that 2X belt frequency approximated 450 CPM and was the likely cause of the high amplitude. A report of this finding was issued on 3/22/89 warning of the high vibration and its possible causes as seen in Figure 2 which is a copy of the **Rank-Ordered Results and Recommendations Report** issued to the client by Technical Associates. Unfortunately, the client did not follow the recommendations given in Figure 2, but simply aligned and tensioned the belts as seen by the 5/23/89 spectrum taken during the next survey. Of course, the 450 CPM belt frequency second harmonic was still present dominating the spectrum. Again a report was issued recommending replacement of the belt followed by proper tensioning and alignment of the motor and fan sheaves.

II. CONCLUSIONS

A. The 450 CPM amplitude was the result of a belt and/or sheave problem.

III. RECOMMENDATIONS

A. Replace the belts which are the dominant vibration source in this machine.

B. Check the fan sheave for excessive runout and repair/replace as needed.

C. Align and tension the belts.

IV. RESULTS OF CORRECTIVE ACTIONS

As seen in the waterfall spectra of Figure 1 and the Position 2H Trend Graph of Figure 3, the alignment and tensioning of the belts reduced vibration significantly, but the 450 CPM vibration remained. Later, the belts were replaced resulting in the spectrum dated January 24, 1990, showing no 450 CPM vibration as noted in the frequency/order/amplitude tabular listing near the middle of Figure 1 that is also dated January 24, 1990. The Trend Graph of Figure 3 shows the improved results which occurred to the Position 2H overall vibration as a result of these corrective actions.

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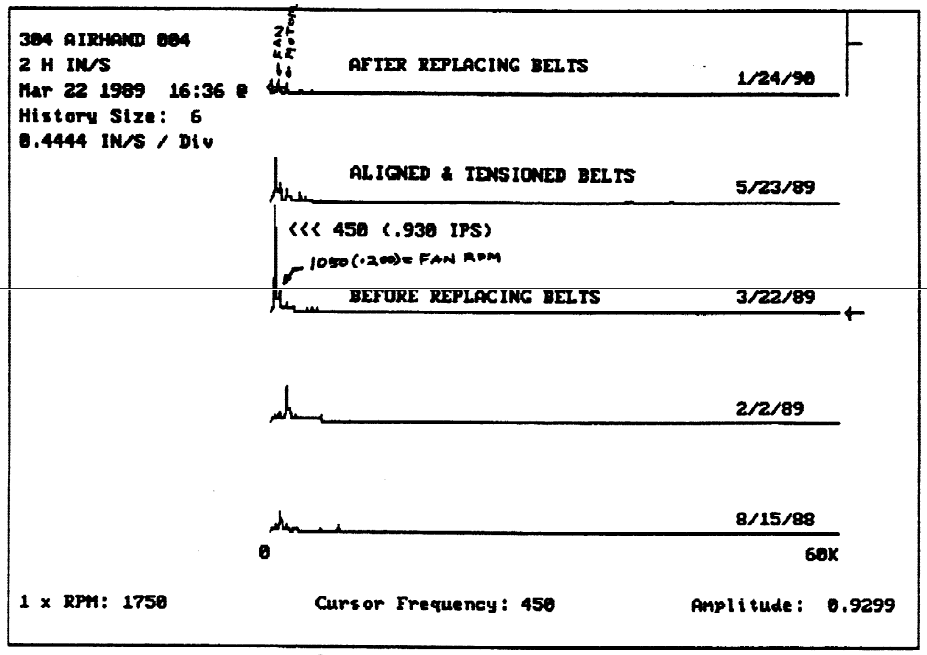
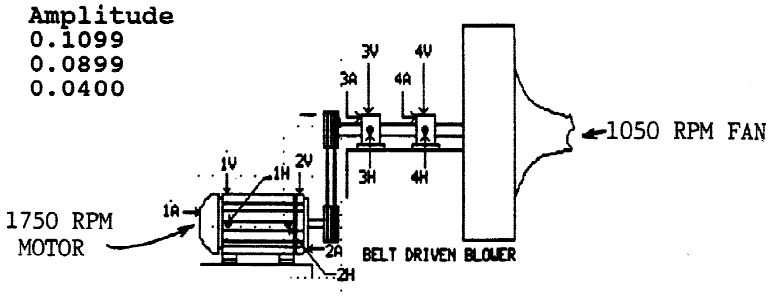


2% x 2.000 = 0.0400 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
304 AIRHAND 004	2	H	IN/S	Jan 24 1990 13:58

Frequency	1 x RPM	Amplitude
900	0.51	0.1099
1800	1.03	0.0899
3300	1.89	0.0400



T - Threshold adjust Q - Quit

FIGURE 1

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SUMMARY OF RESULTS AND RECOMMENDATIONS FOR MACHINES FOUND TO HAVE PROBLEMS

(SORTED FIRST BY PROBLEM PRIORITY, THEN BY PROBLEM SEVERITY WITHIN PRIORITY)

AS OF: 3/22/89

AREA	MACHINE COMPUTER NAME	INDICATED PROBLEM(S): UNBALANCE	MISALIGNMENT/SOFT FOOT	BENT SHAFT	LUBRICATION PROBLEM	BEARING PROBLEM	MECHAN. LOOSENESS	BELT/SHEAVE PROBLEM	RESONANCE PROBLEM	GEAR PROBLEM	BLADE PASSING PROB.	CAVITATION	MOTOR ELECTRICAL	OIL WHIRL/WHIP	LOOSE FOUNDATION	VIB. MONITOR PROB.	SEAL/ROTOR RUB	OTHER (SEE REMARK(S))	*PRIORITY OF PROBLEM	HIGHEST OVERALL VIBRATION: POSITION NO.	LEVEL: m = mils, i = in/sec, g = g's	RECOMMENDATIONS & REMARKS	SEE PLOT NO.
	302 RCPCHIL 001	●																	①	3H	2.20	Extremely high amplitudes measured radially on the motor and chiller with the highest measured at position 3H (2.20 @ 1800 CPM) which indicates severe unbalance. Amplitudes have increased dramatically since September Survey as seen in Plot 37. Unbalance condition is probably caused by a balance weight being thrown off. Recommend shutting compressor off until unit is balanced.	36 37
	304 AIRHAND 004							●											2	2H	1.34	High amplitude measured at position 2H (.929 in/sec) at 1 times belt RPM messured on the motor indicates a belt/sheave problem. Check fan sheave for excessive runout and repair/replace as needed. Align and tension belts.	40
	324 AIRHAND 001	●																	2	2H	1.30	Motor unbalance still present at a slightly higher amplitude as seen in Plot 51. Balance motor as needed.	50 51
	863 RCPOOHP 001														●				2	2H	1.17	Possible loose or inadequate compressor foundation. Inspect compressor base for proper mounting that should include an inertia block or a vibration isolation system. Also, look for any loosened fasteners and tighten properly. Will continue to monitor closely.	89
	302 AIRHAND 001	●	●				●	●											2	4A	1.07	Belt misalignment and/or excessive sheave runout causing a high axial amplitude at position 1A combined with motor unbalance. Mechanical looseness is also indicated. Recommend checking sheaves for excessive runout and repair/replace as needed. Align and tension belts. Tighten all associated hold down bolts. Unbalance condition may be caused by sheave problem, therefore, trend this in future surveys.	33 34 35
	318 COOLTWR 001	●																	2	2H	.913	Motor and/or Drive Shaft Unbalance, likely combined with Drive Shaft Misalignment. Check drive shaft for looseness and runout and correct as necessary.	49

*① = TAKE IMMEDIATE REPAIR ACTION DUE TO SEVERE PROBLEM(S) NOTED. (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
2 = SCHEDULE REPAIRS TO CORRECT PROBLEMS NOTED AT CONVENIENCE. (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
3 = DESIGNATED PROBLEM DOES NOT WARRANT ACTION AT THIS TIME. (CONTINUE ONLY WITH RELIABILITY SURVEY TREND ANALYSIS)

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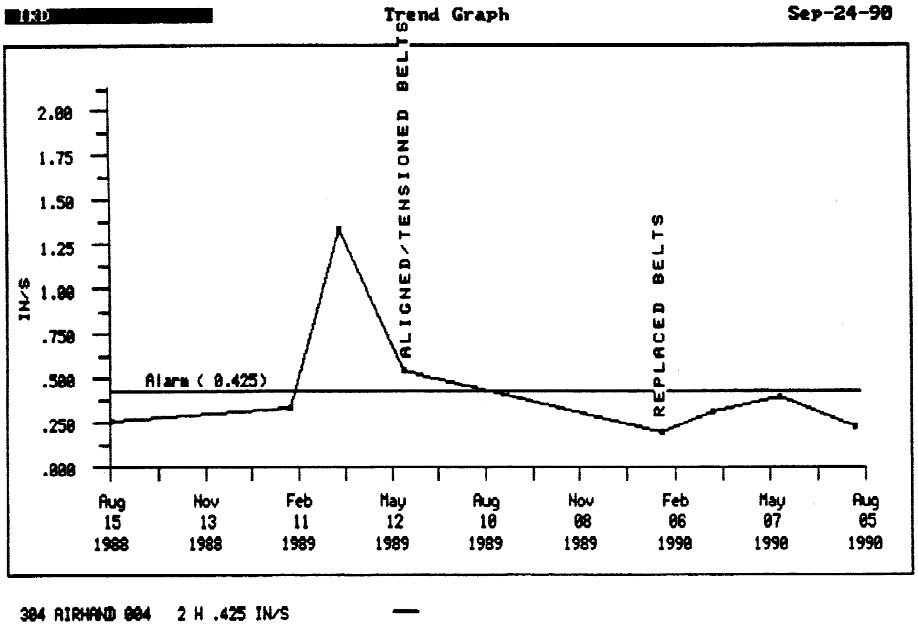
FIGURE 2



IRD Mechanalysis Data Analysis System

Machine 304 AIRHAND 004 Pos 2 Dir H Units IN/S Alarm .425

Ampl	Insp Code	Date mm/dd/yy	Time	Ampl	Insp Code	Date mm/dd/yy	Time
.255	0	08-15-88	10:50				
.335	0	02-02-89	10:09				
1.34	0	03-22-89	16:36				
.537	0	05-23-89	16:54				
.195	0	01-24-90	13:58				
.313	0	03-13-90	14:23				
.396	0	05-15-90	18:00				
.223	0	07-26-90	15:38				
.223	0	07-26-90	15:38				

FIGURE 3

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EFFECT OF TIGHTENING LOOSE BELTS ON MOTOR VIBRATION

I. BACKGROUND

During a Predictive Maintenance Program data collection survey by Technical Associates' Analysts, a belt-driven air handler, 058 AIR HANDLER 002, was noted as having a loose set of belts and an unusual motor vibration of .380 in/sec at 2700 CPM. The .380 in/sec vibration violated the Band 4 alarm of .160 in/sec as seen in Figure 1. A machine diagram showing measurement locations is shown in the upper corner of the spectrum in Figure 1. The motor speed for this unit was nominally 1750 RPM and a hand-held tachometer showed the fan speed was 1530 RPM. A tachometer fan speed measurement is used to identify fan speed due to the possibility of belt slippage and sheave wheel size changes that could affect fan speed.

Since the 2700 CPM frequency was not related to either the motor or fan speeds (1X, 2X, 3X, etc.), another source needed to be considered. The motor and sheave wheel sizes were estimated, along with their center-to-center distance. Then an approximation of the belt frequency as being around 540 CPM was calculated. This identified the 2700 CPM and other frequencies in Figure 1 as harmonics of belt frequency. A Reliability Survey Report was issued reflecting this result. These reports are given after each Reliability Survey giving a continually updated evaluation of condition of each machine included in the plant Predictive Maintenance Program.

II. CONCLUSIONS

A. The high amplitude 2700 CPM frequency was found to be due to a 5X belt frequency vibration. The most likely cause of this high vibration as well as that by the other belt frequency harmonics was concluded to be loose belts.

III. RECOMMENDATIONS

A. Recommendations were made to tighten the loose belts to the proper tension.

B. It was also recommended the mechanic closely inspect both the motor and fan sheave for signs of wear which can contribute to belt looseness problems as well as accelerate wear of the belts themselves.

C. It was also recommended the mechanic closely align the motor sheave with respect to the fan sheave.

IV. RESULTS OF CORRECTIVE ACTIONS

After the belts were properly tightened, the 2700 CPM amplitude was reduced almost 80% from .380 in/sec to .078 in/sec as seen in Figure 2. These types of improvements also result in improved performance and reduced energy costs.

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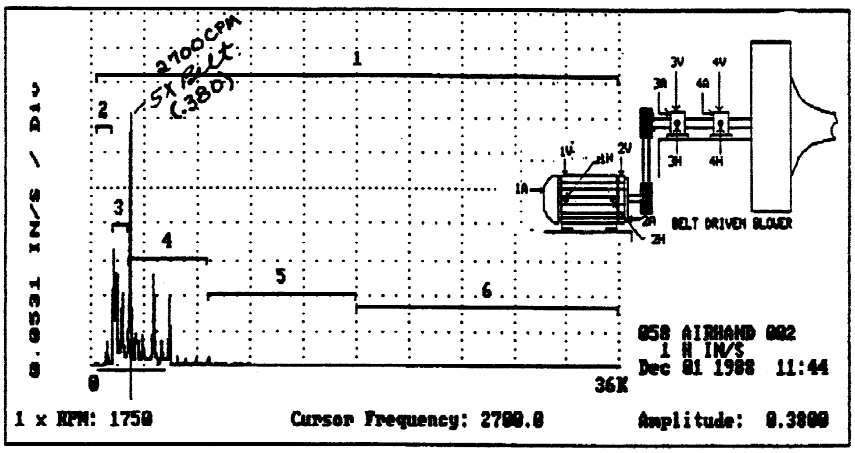

1% x 1.000 = 0.0100 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
058 AIRHAND 002	1	H	IN/S	Dec 01 1988 11:44

Frequency	1 x RPM	Amplitude
1080.0	0.617	0.0349 – 2X Belt.
1530.0	0.874	0.1751 – Fan
1800.0	1.029	0.1400 – Motor
2160.0	1.234	0.1099 – 4X Belt
2700.0	1.543	0.3800 – 5X Belt
3060.0	1.749	0.0500
3510.0	2.006	0.0450
4230.0	2.417	0.1400
4770.0	2.726	0.0349
5310.0	3.034	0.1050
5850.0	3.343	0.0149
6390.0	3.651	0.0100
7110.0	4.063	0.0100
8010.0	4.577	0.0149

T - Threshold adjust Q - Quit

FIGURE 1

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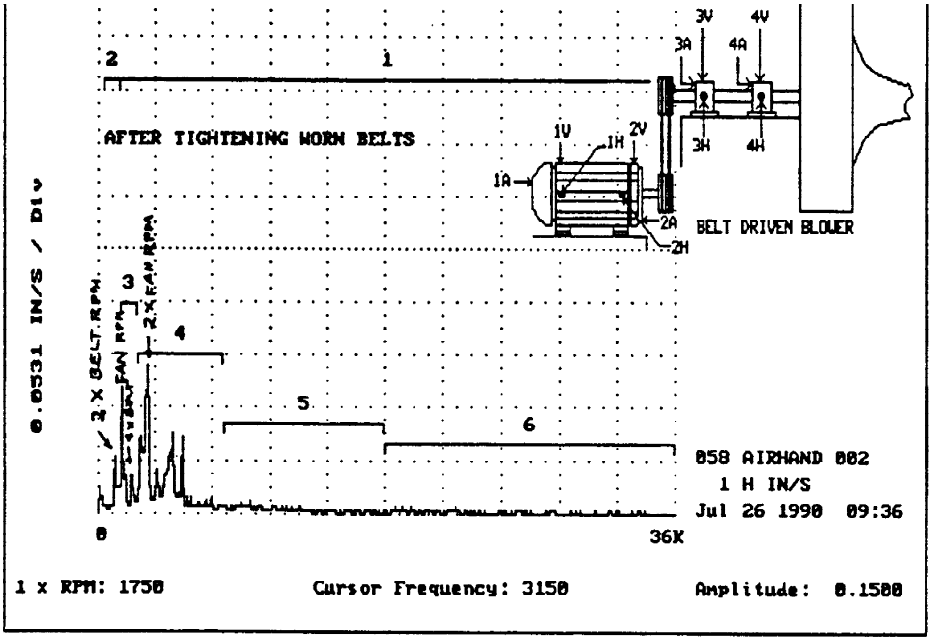

2% x 0.500 = 0.0100 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
058 AIRHAND 002	1	H	IN/S	Jul 26 1990 09:36

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
90	0.051	0.0250	9540	5.451	0.0125
1080	0.617	0.0575 = 2X BELT			
1530	0.874	0.1250 = FAN RPM			
1800	1.029	0.0375 = MOTOR RPM			
2160	1.234	0.0375 = 4X BELT			
2700	1.543	0.0775 = 5X BELT (was .380 in/sec)			
3150	1.800	0.1500 = 2X FAN			
3690	2.109	0.0425			
4680	2.674	0.0800			
5040	2.880	0.0200			
5310	3.034	0.0750			
5580	3.189	0.0150			
5850	3.343	0.0150			
6390	3.651	0.0100			
6840	3.909	0.0125			
7200	4.114	0.0150			
9000	5.143	0.0100			

T - Threshold adjust Q - Quit

FIGURE 2

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